ATTACHMENT A

SUBSTITUTE SPECIFICATION

(Including All Changes From the Specification as filed in Application Serial No. 09/905,254)

Double-Clutch Transmission

Background of the Invention

Field of the Invention

[0001] The present invention relates to a transmission for a motor vehicle. More particularly, the present invention relates to a transmission having multiple shafts, such as a first and a second transmission input shaft and a transmission output shaft, as well as a plurality of gear pairs between the transmission output shaft and the transmission input shafts, including an idler that is arranged on one of the shafts and connected with it, and a fixed drive gear that is non-rotatably arranged on a corresponding shaft so as to mesh with the idler for forming gear steps with different gear ratios between the transmission input shaft and one of the transmission output shafts.

Description of the Related Art

[0002] Such transmissions are known, particularly in connection with internal combustion engines, as well as the separation of the transmission input shafts through a coupling with the crankshaft of the engine and represent the state of the art for the task of further developing and automating these transmissions. One aspect of the task is to manufacture the transmission in the automated version in a cost effective manner. Another aspect of the task is the economical operation of a

drive train with such a transmission. Furthermore, a portion of the task is to provide a method for the ecological and comfortable operation of a motor vehicle with a transmission in accordance with the description herein.

Summary of the Invention

[0003] This task is resolved with a transmission, particularly for motor vehicles, with a plurality of shafts, such as a first and a second transmission input shaft and an output shaft, which can also be formed by two shaft branches that can later be combined by means of a differential or a gear system so that the transmission has at least the following features:

[0004] a) a plurality of gear pairs is arranged between the transmission output shaft and the transmission input shafts, including an idler that is arranged around one of the shafts, respectively, and non-rotatably connected with it, and a driving gear that is non-rotatably arranged on a corresponding shaft so as to mesh with the idler for the purpose of forming gears with different gear ratios between the transmission input shaft and one of the transmission output shafts;

[0005] b) at least one transmission input shaft is driven at least from time to time a drive unit with a drive shaft;

[0006] c) at least one transmission input shaft is connected with a first electrical machine;

[0007] d) the transmission output shaft is connected with at least one driving wheel;

[0008] at least one gear is shifted automatically with an actuator.

[0009] For this embodiment, the idler and fixed gears that are formed by the gear pairs can be arranged on the transmission input shafts and/or the transmission output shaft, wherein it can be beneficial to arrange the idlers on the transmission input shafts. Furthermore, it may be beneficial for other embodiments to arrange the idlers on the transmission output shaft, particularly in transmissions where the appropriate fixed gears can be arranged rationally on the transmission input shafts, because their diameters can be so small that they can be easily manufactured in a firmly-connected version or single-piece version through forging, milling, or stamping processes, through hot-flow processes such as lateral extrusion or similar processes. The drive unit can be formed by an internal combustion engine, for example, a piston engine with a crankshaft, with appropriate devices being provided for damping torsional vibrations, axial, and/or wobble vibrations between the internal combustion engine and the transmission. Furthermore, the drive unit can be formed by a second electrical machine, wherein the first as well as the second electrical machine, which can also be operated in a polyphase manner as an electric motor and/or generator based on the synchronous, asynchronous and/or reluctance principle, and can drive one transmission input shaft, respectively, and can have roughly the same dimensions. Particularly in connection with the usage of an internal combustion engine as the drive unit, it is particularly beneficial to design its drive shaft so that it can be coupled with the transmission input shafts, wherein at least one transmission input shaft, preferably both, can be coupled with the drive shaft. One embodiment in accordance with the invention provides for a design where the clutches are friction clutches, preferably dry friction clutches, in the form

of a double clutch, with this being possibly arranged in the clutch housing of the transmission, i.e. axially between the drive unit and the transmission. For this version, the previously-mentioned damping devices can be integrated in the double clutch. Further, a flywheel can hold the clutches, wherein the various clutch components can be fastened to the flywheel as modules and the flywheel can be a divided flywheel with a two-mass effect.

[0010] Based on one inventive idea, the drive unit can furthermore be an internal combustion engine with a crankshaft that can be connected with one transmission input shaft through a double clutch. For this purpose, an electrical machine – such as the one described above as the first electrical machine – can be connected together with at least one transmission input shaft in such a manner that it can be uncoupled. It can also prove particularly beneficial to arrange the electrical machine so as to allow it to be connected alternatively with both transmission input shafts. This connection can be formed by a friction, shifting, or magnetic clutch, which serves as the connection between the electrical machine and the transmission input shaft by building up electro-magnetic fields, wherein the formation of the connection and/or the control of this clutch can occur through an electric, hydraulic, and/or pneumatic actuator as well as a combined procedure, or in the case of a magnetic clutch through the appropriate control of the electric currents through the device for adjusting the magnetic effect, such as coils or similar devices. Of course, two clutches can also be beneficial for forming an uncouplable connection between the transmission input shafts and the electrical machine,

wherein one clutch can connect the electrical machine with one of the transmission input shafts, and two appropriate actuators can be used for this purpose.

[0011] Beneficial embodiments of the transmission provide for the fact that the transmission output shaft can be arranged basically coaxially to the drive shaft and/or that one of the transmission input shafts is arranged basically coaxially to the drive shaft. It can prove particularly beneficial to arrange one transmission input shaft as a hollow shaft around the other transmission input shaft. In a beneficial version, the gear pairs that form the individual gears can be arranged alternating on the two transmission input shafts in dependence on the gear ratios. This way it is possible to operate the vehicle through a transmission input shaft that is connected with the internal combustion engine through the appropriate clutch, and a gear with one gear ratio, while on the other transmission input shaft the next gear ratio is engaged with a disengaged clutch between the transmission input shaft and the internal combustion engine. This way, for example, four, preferably six, separate forward speeds and one reverse speed can be incorporated in these transmissions, wherein those gears with an increasing gear ratio can be arranged on one transmission input shaft and those gears with gear ratios in between the gear ratios of those on the first transmission input shaft can be arranged on the other transmission input shaft. The reverse gear can be arranged on either of the two transmission input shafts. In an alternative embodiment, the motor vehicle can be operated purely electrically in reverse, wherein the electrical machine is operated in the opposite rotational direction.

[0012] The preferred starting gear with the smallest gear ratio can, for example, be arranged on the first input shaft, the second gear with the next higher gear ratio on the second input shaft, the third gear again on the first input shaft and the fourth gear again on the second input shaft, etc. The electrical machine can be connected with the transmission input shaft that contains the gear with the smallest or the gear with the next smaller gear ratio. The individual gears are preferably formed through fixed gears and idlers, which are arranged on one shaft, such as the transmission input shaft and the transmission output shaft, respectively, wherein for the purpose of activating the gear the appropriate idler is connected with the shaft, for example, through a sliding sleeve. In a beneficial version, the idlers can be arranged on the transmission input shafts, on the transmission output shaft or, depending on the requirement, alternating on one of the transmission input shafts and the transmission output shaft or driven shaft. As is known, the idlers can be placed on the appropriate shafts such as transmission input shaft and/or transmission output shaft in a synchronized manner with regard to a speed between the shafts carrying the two gear pairs, wherein this synchronization process can occur with conventional synchronizing devices or alternatively, or additionally, with the electrical machine, and wherein the electrical machine can be used in a driving or braking manner in accordance with the required minimization of the differential speed between the two shafts for achieving a synchronizing speed. Furthermore, it may be particularly beneficial to accelerate the synchronization process by decelerating or accelerating the transmission input shaft, by operating it at least in a slipping manner with the drive unit through the clutch, which is generally

disengaged when the torque is transferred through the other transmission input shaft.

[0013] With regard to the arrangement of the electrical machine on the transmission, it has proven particularly beneficial to arrange it on the end of the transmission input shaft that is opposite the drive unit, such as the internal combustion engine. Of course, it can also be beneficial to arrange the electrical machine parallel to one of the transmission input shafts, wherein an arrangement parallel to the axis through an active connection such as a belt, chain, toothed wheel connection or the like is selected, and the electrical machine can be arranged in the area of the double clutch or at the axial height of the gear. When arranged on the side of the gear that is opposite the drive unit, a coaxial arrangement of the electrical machine to the transmission input shaft, with which the electrical machine is connected, can be beneficial. Additionally, the electrical machine can be arranged around the clutch, for example, around the double clutch of the double clutch transmission, which offers the advantage that additional axial space is largely eliminated and that due to a larger diameter the electrical machine can have a stronger, i.e. more powerful, design. With regard to an active connection of the electrical machine to the double clutch gear, it may be beneficial to arrange the electrical machine on the gears that are arranged on one of the transmission input shafts, next to the direct coupling to the transmission input shaft. This way, the electrical machine, when used as a motor or generator, can be adjusted to the speed-dependent maximum efficiency level of the electrical machine by utilizing the various gear ratios of the gears that can be carried on the transmission input shaft.

On the other hand, it has turned out that particularly during recouperation processes, kinetic energy that can be converted into electrical energy is recuperated over a long power path, such as in the case of an operative connection of the electrical machine with the gear with the largest gear ratio, when recuperating in a gear with small gear ratio. In this case, kinetic energy is guided, for example, through three gear pairs so that a loss of efficiency must be tolerated. Based on the inventive idea it can prove particularly beneficial in such cases to actively connect the electrical machine preferably on the gears with a mean gear ratio, for example, in dependence on the selection of the transmission input shaft, preferably on gear II or gear IV and/or gear III.

[0014] Additionally, the electrical machine can be arranged on the transmission output shaft, wherein it is rotatably arranged and can be operatively connected with the transmission input shaft. This is particularly beneficial in so-called in-line transmissions where the transmission output shaft is arranged coaxially to the crankshaft. The electrical machine can be positioned on the transmission output shaft at the end of the transmission that is opposite the crankshaft, and can thus be arranged optimally for spatial reasons. This can include a rotatable arrangement around the transmission output shaft so as to arrange the rotor around the transmission output shaft and rotatably support it, or to rotatably support the rotor opposite to the transmission housing. In both cases, the stator must be firmly connected with the transmission housing. As in the remaining embodiments, the electrical machine can have an external or internal design, i.e. with a rotor that is arranged around the stator or within the stator. The electrical

machine can basically be a synchronous, asynchronous, or reluctance type. The operative connection between rotor and transmission input shaft can occur through a belt drive, a geared connection, or the like, wherein it may be particularly beneficial to connect the electrical machine with a gear of a pair of gears, for example, a gear step with high gear ratio, e.g., gear V. Beneficially, a gear step is selected whose gear ratio is larger than the gear ratio of the direct drive rotational speed of the crankshaft and is equal to the rotational speed of the transmission input shaft, so that an appropriate gear ratio of these two gear steps between the electrical machine and the internal combustion engine exists, which allows the electrical machine to be operated in a generator mode at efficient rotational speeds and the internal combustion engine can be started by the electrical machine with appropriate lower rotational speeds of the crankshaft and high rotational speeds of the electrical machine. Additionally, it may be beneficial to rotatably connect the rotor, for example, with the transmission output shaft through a clutch to further spread the rotational speed range of the electrical machine.

[0015] At least one auxiliary unit can be drivingly connected with the electrical machine, and it may be particularly beneficial when in the case of an arrangement of the electrical machine parallel to the axis of the transmission input shaft the electrical machine is integrated into the belt pulley side of the auxiliary unit. The electrical machine can perform a drive function in the conventional sense, wherein the electrical machine beneficially can be uncoupled from the transmission input shaft so that the auxiliary units can be operated by the electrical machine independently of the rotational speed of the transmission input shaft, i.e., also

independently of the rotational speeds of the driving wheels and the rotational speed of the drive shaft of the internal combustion engine. This offers the advantage that, if it is desired to operate the auxiliary units electrically independently of the drive unit, the separate supply of these auxiliary units with one electrical machine, respectively, can be eliminated and a corresponding weight saving can result. Furthermore, a gear ratio can be provided between the electrical machine and at least one auxiliary unit that can be adjusted variably, for example, through a variably adjustable transmission having an endless torque-transmitting means (CVT), or through geared connections that can be actuated automatically or manually. It can also prove beneficial to uncouple the electrical machine from at least one auxiliary unit through a so-called auxiliary unit clutch. Several auxiliary units that are arranged on a belt pulley side can be separated from, connected with and/or have a gear ratio in relation to each other and/or to the electrical machine, which can be accomplished with clutches, free-wheels and appropriate transmissions for selecting variable and/or fixed gear ratios.

[0016] Based on another idea of the invention, the connection between the drive shaft and at least one of the transmission input shafts can be either geared down or geared up. This transmission ratio or input transmission ratio can occur beneficially through gear steps, wherein a graduation of the transmission input shafts between each other can occur by subjecting one transmission input shaft to a gear ratio, but not the other. Additionally the r.p.m. range of the transmission input shafts can be adjusted so as to operate the electrical machine independently from the engaged gear at an optimized rotational speed, i.e., a speed that has been

adjusted for the electrical machine with regard to its efficiency. Of course, the corresponding transmission ratio can also occur directly between the transmission input shaft and the electrical machine, particularly in the case of an arrangement of the transmission input shaft parallel to the axis of the electrical machine with an operative connection between these components, such as a belt drive, a chain drive, a gear drive, and the like.

[0017] Based on another idea of the invention, the drive train consisting of the drive unit, such as an internal combustion engine, the clutch unit, such as a double clutch, and the transmission, such as the double-clutch transmission, is provided for automatic operation, wherein at least one clutch and/or one gear step can be engaged automatically in dependence on the driving situation. The design of the drive train as a fully automatic transmission with two clutches that can be actuated fully automatically, and the fully automatic actuation of all gears, however, is advantageous. This way, at least one gear step or one clutch is actuated by an actuator, which can be an electric, hydraulic, pneumatic or combined actuator. In a beneficial embodiment such an actuator is provided for each gear step, wherein it can prove particularly beneficial to engage two neighboring gears, respectively, that are arranged on one transmission input shaft through shift sleeves such as sliding sleeves that are engaged by an appropriate actuator, for example, a pair of gears consisting of a first gear and a neighboring gear on the transmission input shaft can be engaged by an actuator. A pair of gears can be formed, for example, by the first and the third gear steps, wherein the shift sleeve can override a possibly adjustable neutral position between activation of the first gear and activation of the second

gear by forming a positive lock with the transmission input shaft. It can be beneficial to combine an individual gear that cannot be combined with a pair of gears with the transmission input shaft through connection with the electrical machine so that with this sliding sleeve an actuator either engages this gear or connects the electrical machine with the transmission input shaft or activates optionally a neutral mode.

[0018] Based on another idea of the invention, gear ratio steps can be actuated through an actuator on the transmission input shafts that are equipped with a synchronizing device on the last gear pair, for example, the first transmission input shaft that is not operatively connected with the electrical machine, wherein gear ratio steps are engaged by connecting an idler with the shaft that holds it through an end output element, which is part of an end output mechanism that is actuated by the end actuating mechanism, and wherein the shifting sequence of the gear ratio steps is not set in the end actuating mechanism. The end output element here is the element that is moved in order to set a gear ratio, i.e., the one that establishes the connection between two power transmission devices, for example, a clutch sleeve. This end output element is part of the end output mechanism, which apart from the clutch sleeve comprises a shift fork, for example, that is connected with the clutch sleeve and can be moved with a shift finger that can be actively connected with it, causing the clutch sleeve to be moved in order to engage or disengage a gear ratio step, wherein the shift finger is part of the end actuating mechanism that actuates the end output mechanism. The end actuating mechanism, which can be controlled by an actuator and can include the kinematic transmission of the actuator movement onto an actuating element, such as a shift

finger, can comprise at least a main actuating element such as a shift finger which is operatively connected with the end output mechanisms, such as shift forks and sliding sleeves, in such a way that a gear ratio step can be engaged and that at least one main actuating element can be actively connected with another end output mechanism without having to disengage the previously engaged gear ratio step, wherein the end actuating mechanism can comprise at least one auxiliary actuating element, for example, at least one additional shift cam. The end output mechanisms in accordance with the invention can comprise connecting elements such as shift forks, which are equipped with a first functional area for engagement of a main actuating element and a second functional area for engagement of an auxiliary actuating element. The auxiliary actuating element can be arranged, for example, on a selector shaft that rotates around its longitudinal axis upon actuation, wherein the second functional area can be designed so as to allow power to be transmitted - upon rotation of the selector shaft - from one auxiliary actuating element to the second functional area in the disengaging direction of the associated gear ratio step, with this power being equal to or larger than the force that is required for disengagement.

[0019] As soon as at least one main actuating element operatively connects with an end output mechanism, at least one auxiliary actuating element can actively connect with at least one additional end output mechanism. For the purpose of disengaging the appropriate gear ratio steps, it can prove furthermore beneficial to actuate another end output mechanism through at least one auxiliary actuating element while actuating an end output mechanism for engaging a gear ratio step

through at least one main actuating element. The end actuating mechanism can be designed so as to allow only one gear ratio step of a transmission input shaft to be engaged at one time. Furthermore, auxiliary actuating elements and the functional areas in the end output mechanisms can interact in such a way that a gear ratio step is disengaged when rotating the selector shaft regardless of the rotational direction, wherein an auxiliary actuating element and these functional areas are of symmetrical design. It is beneficial when at least one auxiliary actuating element has two cam-like end areas and the functional areas have corresponding recesses. Furthermore, the functional areas can be equipped with two cam-like end areas and at least one auxiliary actuating element can have corresponding recesses. Transmission of power between the auxiliary actuating element and the functional areas can occur, for example, through the tips of the cam-like end areas or through the side surfaces of the cam-like end areas. Further embodiments and a more detailed description with regard to function are revealed in the unpublished application DE 101 08 990.2, which hereby is included with its full content in the present application.

[0020] Based on the idea of the invention, the end actuating mechanism can also perform the synchronization of the transmission input shaft through the synchronizing device on the last gear pair, for example, with the sliding sleeve actuating only the friction device of the synchronizing device of the last gear pair by performing an axial movement that corresponds to the engagement of the last gear ratio step, however not ultimately actuating the clutch, but rather moving it back into the starting position after decelerating the transmission input shaft and the

transmission input shaft reaching the synchronizing speed. Of course, the friction device of the synchronizing device on the last gear pair is appropriately designed so as to perform the synchronization of the transmission input shaft for all gear ratio shifts on the gear ratio steps that are arranged on this transmission input shaft. This feature can include particularly resistant, wear-proof friction components such as ceramic friction disks or conventional friction linings with a large wear range. Furthermore, it can provide for easily exchangeable friction disks, e.g. disks that are open on one side and arranged in disk cages, which can be slid easily over the shaft around which the synchronizing device is arranged. The usage of the end actuating mechanism with at least one main and one auxiliary actuating element can be particularly beneficial in that the synchronizing device on the last gear pair is actuated through the auxiliary actuating element for the purpose of synchronizing the rotational speed of the first transmission input shaft to the speed of the transmission output shaft during a gear ratio step switch and that the gear ratio step change occurs through the main actuating element. This allows the deceleration of the transmission input shaft through the auxiliary actuating element to occur nearly simultaneously and in the same operation as the disengagement of the engaged gear ratio step with the main element, so that practically no loss of time occurs over the arrangement of separate synchronizing devices on each gear step – an arrangement that requires considerably more space and is more cost intensive – and a simplified actuation of the synchronizing device over actuation through a separate or kinematically complicated end actuating mechanism that is dependent upon the end actuating mechanism for disengaging and engaging the remaining

gear steps can be suggested. Of course, the auxiliary actuating element can also engage the last gear ratio step.

[0021] Based on an idea of the invention, the drive unit can furthermore be an internal combustion engine with a crankshaft, which can be connected with a transmission input shaft through a double clutch. For this, an electrical machine – as the one described above as first electrical machine – can be connected additionally with at least one transmission input shaft in such a manner that it can be uncoupled. It may be particularly beneficial to arrange the electrical machine in such a way that it can be connected alternatively with both transmission input shafts. This connection can be formed with a friction, shift, or magnetic clutch, which creates the connection of the electrical machine with the transmission input shaft by creating electro-magnetic fields, wherein the formation of the connection and/or the control of this clutch can occur through an actuator with electric, hydraulic and/or pneumatic as well as combined features or in the case of a magnetic clutch by the appropriate control of electric currents by devices that adjust the magnetic effect, such as coils or the like. Of course, two clutches may also be advantageous for forming an uncouplable connection between the transmission input shafts and the electrical machine, wherein one clutch from time to time can connect the electrical machine with one of the transmission input shafts and for which two appropriate actuators can be employed.

[0022] At least one auxiliary unit can be connected with the electrical machine from a drive point of view; it may be particularly beneficial if in the case of an arrangement of the electrical machine parallel to the axis of the transmission

input shaft the electrical machine is integrated into the belt pulley side of the auxiliary unit. The electrical machine can perform a drive function in the conventional sense, wherein the electrical machine in a beneficial embodiment can be uncoupled from the transmission input shaft so that the auxiliary units can be operated by the electrical machine independently from the rotational speeds of the transmission input shaft, i.e., also independently from the rotational speeds of the driving wheels and the rotational speed of the drive shaft of the internal combustion engine. If it is desired to operate the auxiliary units electrically independently from the drive unit, this design is beneficial because it eliminates the separate supply of these auxiliary units with an electrical machine and reduces the weight accordingly. Furthermore, a gear ratio may be provided between the electrical machine and at least one auxiliary unit that can be adjusted variably, for example, through a variably adjustable endless belt transmission (CVT) or through gear connections that can be actuated automatically or manually. It can also be beneficial to uncouple the electrical machine from at least one auxiliary unit through a so-called auxiliary unit clutch. Several auxiliary units that are arranged on a belt pulley side can be separated from, connected with and/or subjected to a gear ratio process with each other and/or the electrical machine, also through clutches, free-wheels and appropriate transmissions for selecting variable and/or fixed gear ratios.

[0023] Another beneficial embodiment may involve the usage of energy recovered during recouperation for supplying hydraulic accumulators when employing hydraulic devices, e.g., a hydraulic actuating device of at least one of the clutches, wherein, for example, the recouperation energy that has been converted

into electrical energy supplies an electric pump or wherein a pump that is coupled to the drive train directly supplies the accumulator during a recuperating process by utilizing the kinetic energy that is provided by the driving wheels. The advantage of such methods and designs is that a usually occurring intermediate storage process, for example, in an electric battery, can be avoided, thus allowing the overall efficiency of the recuperating process and thus of the motor vehicle to increase. Of course, the direct operation of a auxiliary unit with kinetic energy offers the greatest efficiency, and in particular applications, for example when an auxiliary unit cannot be directly operatively connected with the drive train for spatial reasons, the energy created by the electrical machine can be supplied directly, and without intermediate storage in a battery, to an electrically operated auxiliary unit, e.g., a pump for an actuating device for clutches, power steering devices, chassis stabilization devices, and/or the like, a compressor for air conditioning devices, devices for compression of the intake air for the internal combustion engine, for compressed air brakes and/or the like, which can be arranged in accordance with the spatial conditions. Prioritization of the power supply and/or energy supply with a combination of individual power users and/or energy consumers can be provided in dependence of the charge state of the electric accumulator during a recouperation process. The highest priority is beneficially assigned to the supply of safety-relevant users such as the power steering pumps, the braking devices, the actuating devices for clutches, the chassis stabilizing components, engine controls and the like before users providing comfort such as air conditioning compressors, seat heating, window openers and the like; after that excess energy can be stored as electrical energy in

an electric battery, or e.g. as thermodynamic energy in an air conditioning compressor, e.g. as dry ice, as condensed supercritical gas, or the like.

[0024] In the shifting device mentioned above in accordance with the invention, the end output element is the element that is moved in order to set the gear ratio, i.e., the one which establishes the connection between two power transmission devices, such as a clutch sleeve. This end output element is part of the end output mechanism, which apart from the clutch sleeve comprises, e.g., a shift fork, which is connected with the clutch sleeve and can be pushed by a shift finger that is operatively connected with it, so that the clutch sleeve is moved in order to engage or disengage a gear ratio step, wherein the shift finger is part of the end actuating mechanism that actuates the end output mechanism; the end actuating mechanism is the entire kinematic chain between the shift and/or selection input and the end output mechanism.

[0025] In state-of-the-art transmissions, the end output mechanism and end actuating mechanism interact so as to allow a gear ratio step to only become engaged when no other gear ratio step has been engaged. In order to engage a gear ratio step, all other gear ratio steps must therefore be disengaged first. For example, shift fork openings, with which the shift finger can be connected in order to shift the clutch sleeve through the respective shift fork, are designed in such a way that the shift finger can only connect with another shift fork if the clutch sleeve, with whose shift fork it is connected at that time, has assumed a neutral position. With regard to a conventional manual transmission with an H-shift pattern, this means that a selection movement of the gear shift lever from one shift passageway into

another can only occur in the neutral passageway, wherein with a lever movement from one shift passageway into the neutral passageway the gear ratio step that was just engaged will always be disengaged. The gear ratio steps, which can be changed with the same clutch sleeve, cannot be engaged simultaneously anyhow. Therefore, it is necessary for a shift process to disengage a previous gear ratio step, perform a selection movement and then engage a new gear ratio step; during this time, the flow of torque is interrupted by a disengaged drive clutch since the branch must be free from load during the shifting process.

[0026] Particularly in the case of transmissions that can be shifted under load, where the gear ratio steps form groups between which tractive force-uninterrupted load shifting can be performed, for example, by allowing the gear ratio steps to be included in various parallel transmission branches that are associated with different output elements of a friction clutch, so that a continuous change of the torque from one branch to the next can be effected by actuating a friction clutch, designs of connections of the end output mechanism with the end actuating mechanism have been known that permit the engagement of one gear ratio step without having to disengage another gear ratio step that has possibly already been engaged. This way it is possible to engage several gear ratio steps simultaneously in several transmission branches through a single end actuating mechanism by first engaging a gear ratio step in one branch, with the shift finger then connecting with other shift forks in order to engage additional gear ratio steps without having to disengage the gear ratio step in question. In this connection, reference is made to

application DE 100 20 821 A1 by the applicant, whose contents are also part of the disclosure content of the present application.

[0027] Generally, two groups of gear ratio steps are formed, wherein with regard to the graduation of their gear ratio successive gear ratio steps are part of different groups. For example, in the case of a manual transmission with one reverse gear (R) and six forward gears (I-VI) one group comprises gears I, III and V and the other group comprises gears R, II, IV and VI.

[0028] Such a transmission offers the possibility of having engaged a gear ratio step in the transmission branch that is closed in the flow of torque through the friction clutch, and to then engage – in another still open branch – the gear ratio step into which subsequently the switch is supposed to occur through diversion of the flow of torque into the appropriate branch. During an acceleration process, for example, while the gear III is engaged in a closed transmission branch, the gear IV can be engaged in another branch. If, however, suddenly a shift back into gear II should occur, the gear IV must first be disengaged and then the gear II engaged, which represents a particularly high time delay when the gears II and IV are shifted by different clutch sleeves.

[0029] It is also conceivable to have a situation where in the open transmission branch more than one gear ratio step is engaged, which represents a very large safety risk because as soon as this branch is connected into the flow of torque several gear ratio steps with differing gear ratios become operative, which can block or even destroy the transmission.

[0030] Additionally, so-called drum controller transmissions are known, where the end output mechanisms of the gear ratio steps are actuated through a rotatable drum controller. In the drum controller, e.g. shift-gate-like grooves are incorporated, which extend on the surface of the cylindrical drum controller both in the circumferential direction and in the axial direction, so that upon rotation of the drum controller around its longitudinal axis shift forks, which are connected with the drum controller kinematically through elements sliding in the grooves, move in the axial direction of the drum controller; the shifting sequence of the gear ratio steps in relation to the rotation of the selector shaft is set by the course of the grooves. With an appropriate design of the grooves, such drum controller transmissions allow the disengagement of an old and the engagement of a new gear ratio step to overlap, which offers a certain time advantage during the shifting process and thus reduces the duration of the tractive force interruption, however only sequential shifting is possible, e.g., shifting from gear I into gear III is just as impossible as a direct shifting back from, e.g., gear V into gear I.

[0031] This problem is solved with the feature that in a transmission where the end actuating mechanism comprises at least one main actuating element such as shift fingers that interact, e.g., through axial displacement of a selector shaft on which it is arranged, with the end output mechanisms, which are formed, e.g., by shift forks and clutch sleeves that are operatively connected with them, in such a manner that a gear ratio step can be engaged, e.g., by rotating the selector shaft, on which at least one main actuating element is arranged, and that it can then operatively connect with another end output mechanism without having to

disengage the previously engaged gear ratio step. The end actuating mechanism comprises at least one auxiliary actuating element.

[0032] In accordance with one particularly preferred embodiment at least one auxiliary actuating element interacts with at least one additional end output mechanism, e.g., in a certain position, a main actuating element interacts with an end output mechanism, while simultaneously auxiliary actuating elements interact with the additional end output mechanisms, as soon as at least one main actuating element interacts with the end output mechanism. Upon actuation of an end output mechanism for engaging a gear ratio step through at least one main actuating element, e.g., by rotating the selector shaft, it is beneficial if at the same time at least one additional end output mechanism is actuated through at least one auxiliary actuating element for disengaging the appropriate gear ratio steps. It is particularly useful that this way only one gear ratio step can be engaged at one time, and that due to the overlapping disengagement of the old and engagement of the new gear ratio step, as well as the already performed selective movement, a considerable time advantage is achieved.

[0033] Based on another, also particularly preferred embodiment for a transmission where the gear ratio steps form groups, among which a tractive-force-uninterrupted change can occur, at least one auxiliary actuating element is operatively connected with at least one additional end output mechanism of the same group as soon as at least one main actuating element interacts with an end output mechanism of a group. In this embodiment, it is very useful that upon actuation of a end output mechanism of one group for engaging a gear ratio step

through at least one main actuating element at the same time that at least one additional end output mechanism of the same group is actuated through at least one auxiliary actuating element for disengaging the appropriate gear ratio steps. It is beneficial if at least one auxiliary actuating element interacts with no end output mechanism of the other group as soon as at least one main actuating element is operatively connected with an end output mechanism of one group. It is very useful that this way a gear ratio step can be engaged simultaneously in each group, but not several gear ratio steps of one group.

[0034] Based on an exemplary, but particularly preferred embodiment of the end output mechanisms, which comprise connecting elements such as shift forks, the end output mechanisms are equipped with a first functional area for engaging a main actuating element and a second functional area for engaging an auxiliary actuating element so that each end output mechanism can be actuated through a main actuating element or through an auxiliary actuating element. On a transmission at least one auxiliary actuating element is here arranged on the selector shaft that can rotate around its longitudinal axis upon actuation, and the second functional area is designed in such a way that upon a rotation of the selector shaft a force can be transmitted from one auxiliary actuating element to the second functional area in the disengagement direction of the appropriate gear ratio step, with this force being equal to or larger than the force that is required for disengagement. The connection between the auxiliary actuating element and the end output mechanism must not be suited to also transfer a force for engaging a gear ratio step.

[0035] In another embodiment a design of at least one auxiliary actuating element is preferred that makes it possible to connect the auxiliary actuating element with at least two end output mechanisms. For this, at least one auxiliary actuating element is particularly wide in the selector shaft axial direction, which preferably corresponds at least roughly to the width of two shift fork openings and their joint spacing.

[0036] Based on a particularly preferred embodiment, at least one auxiliary actuating element and the second functional areas operate together so as to disengage a gear ratio step upon rotation of the selector shaft independently from the rotational direction. Starting from the original position in which the selector shaft is in a mean position in relation to its rotation and in which also the main actuating element has become engaged with the first functional area of a end output mechanism, a gear ratio step is engaged by rotating the selector shaft either to the right or the left, wherein in any case the at least one auxiliary actuating element actuates the gear ratio step(s) that is (are) are allocated to it with regard to disengagement.

[0037] In the embodiment it is considered particularly beneficial if for this purpose at least one auxiliary actuating element and the second functional areas are of symmetrical design.

[0038] In a particularly preferred example, at least one auxiliary actuating element is equipped with two cam-like end areas and the second functional areas with corresponding recesses.

[0039] In another, also particularly preferred embodiment, the second functional areas are equipped with two cam-like end areas and the at least one auxiliary actuating element with corresponding recesses.

[0040] Transmission of power between the auxiliary actuating element and the second functional area occurs through the tips of the cam-like end areas, wherein in another embodiment it is also useful if the transmission of power between the auxiliary actuating element and the second functional area occurs through the side surfaces of the cam-like end areas.

[0041] In order to solve the problem, these beneficial embodiments are based on an inventive method that contains at least the following procedural steps:

[0042] the drive unit drives at least one of the transmission input shafts at least some of the time;

[0043] the first electrical machine drives one of the transmission input shafts at least some of the time;

[0044] the first electrical machine is driven by one of the transmission input shafts at least some of the time.

[0045] The method in accordance with the invention can at least provide for a starting of the drive unit, which has the design of an internal combustion engine, wherein in the case of a cold internal combustion engine this unit is started preferably through a method that uses the idea of the invention in connection with beneficial arrangements of the drive train that include in each case one clutch between the internal combustion engine and the transmission input shaft and contain the following procedural steps:

[0046] both clutches are disengaged;

[0047] no gear has been engaged between the first transmission input shaft, with which the first electrical machine is connected from a drive point of view, and the transmission output shaft;

[0048] a gear with preferably a small gear ratio or gear reduction has been engaged between the second transmission input shaft and the transmission output shaft;

[0049] the first electrical machine is driving the first transmission input shaft;

[0050] the clutch in the power distribution flow between the first transmission input shaft and the drive shaft is disengaged after reaching the torque that is required for a cold start of the drive unit;

[0051] after starting the drive unit the clutch in the power distribution flow between the drive shaft and the second transmission input shaft is engaged and the vehicle starts to run.

[0052] This method can alternatively or additionally be combined with another method for starting the internal combustion engine, wherein this method is preferably employed for a drive unit in the warmed-up state and contains the following procedural steps:

[0053] no gear has been engaged between the first transmission input shaft, with which the first electrical machine is connected from a drive point of view, and the transmission output shaft;

[0054] a gear with preferably a small gear ratio or gear reduction has been engaged between the second transmission input shaft and the transmission output shaft;

[0055] the clutch in the power distribution flow between the first transmission input shaft and the drive shaft is engaged;

[0056] the first electrical machine is being driven and the drive unit is started;

[0057] by engaging the clutch in the power distribution flow between the drive shaft and the second transmission input shaft the vehicle starts to run.

[0058] Alternatively or additionally, the following starting procedure can prove beneficial, particularly for a cold internal combustion engine in connection with the arrangement of a drive gear on the first transmission input shaft and an idler with a shift sleeve, a so-called triplex sleeve, that interacts with the drive gear, with the sleeve being arranged on the transmission output shaft and being able to connect selectively the gears of a gear pair with each other, to connect one of the gears positively with the transmission output shaft or to assume a neutral position without a connecting function:

[0059] no gear is engaged between the first transmission input shaft, which is connected with the first electrical machine from a drive point of view, and the transmission output shaft;

[0060] the two gears are connected with each other through the triplex sleeve between the second transmission input shaft and the transmission output shaft;

[0061] the clutch in the power distribution flow between the second transmission input shaft and the drive shaft is engaged;

[0062] the electrical machine is being driven and the drive unit is being started;

[0063] the clutch between the drive unit and second transmission input shaft is being disengaged;

[0064] the second transmission input shaft and the transmission output shaft are decelerated to a negligible rotational speed, for example, through the electrical machine;

[0065] the triplex sleeve is moved into the neutral position;

[0066] a gear with a small gear ratio between the second transmission input shaft and transmission output shaft is engaged;

[0067] by engaging the clutch in the power distribution flow between the drive shaft and the second transmission input shaft the vehicle starts to move.

[0068] A large advantage of this method is a cold start of the internal combustion engine at high rotational speed – caused by the gear ratio steps of two gears, for example, the second and the fifth gears – and thus a reduced torque of the electrical machine. In connection with an appropriate transmission design, the elimination of an impulse start with a cold internal combustion engine is made possible, also and particularly for possible temperatures under the freezing point, and the electrical machine can become more cost effective and have smaller torques. This can lead to enormous cost and space savings.

[0069] The method in accordance with the invention can additionally comprise the following procedural steps for operating the first electrical machine as a generator for creating electrical energy:

[0070] the first electrical machine is driven by the drive unit or, for a driving mode such as recouperation, by at least one driving wheel;

[0071] when driven by the drive unit one of the two clutches in the power distribution flow between the drive shaft and one transmission input shaft is selectively engaged;

[0072] when driven by at least one driving wheel both clutches are disengaged,

[0073] wherein it may be beneficial to operate the electrical machine in dependence on the charge state of electrical energy accumulators, such as a high current battery, a power capacitor, and/or the like, i.e. to connect it with the transmission input shaft, which thus transmits a torque, which is transmitted from the wheels and/or from the drive unit to the shaft, to the electrical machine.

[0074] For the method in accordance with the invention, the following torque flows can be beneficial:

[0075] torque is transmitted from the drive shaft of the drive unit through the engaged clutch in the power distribution flow between the first transmission input shaft, with which the electrical machine is operatively connected, and the drive shaft to the first transmission input shaft and from there to the rotor shaft of the electrical machine;

[0076] torque is transmitted from the drive shaft of the drive unit through the engaged clutch in the power distribution flow between the second transmission input shaft without electrical machine through a pair of gears to the transmission

output shaft, from there through a pair of gears to the first transmission input shaft and from there to the rotor shaft of the electrical machine;

[0077] torque is transmitted from at least one driving wheel to the transmission output shaft and from there through a pair of gears via the first transmission input shaft to the rotor shaft of the first electrical machine. The first electrical machine can be operated at a rotational speed, preferably by selecting an appropriate gear pair between the transmission output shaft and the first transmission input shaft, where it reaches its optimal operating point with regard to efficiency. It may be beneficial to uncouple the drive unit from the first transmission input shaft during the recouperation process with a switch from "pull" to "push" by disengaging the clutch between the first transmission input shaft and the drive shaft in a delayed fashion, e.g., with a delay of > 0.3 seconds after the switch from "pull" to "push."

[0078] The recouperation process can be performed particularly beneficially in connection with a transmission with an electrical machine that can be connected between the transmission input shafts because a switch of the electrical machine to the appropriate transmission input shaft and selection of the most favorable gear for the recouperation process allows efficiency to be improved further since all gears of the transmission can be used for adjusting the rotational speed with the highest efficiency of the electrical machine. Based on another inventive idea, recouperation energy can be stored through additional energy storage types, e.g. thermal energy, pressure, and the like, particularly in the case of an already charged electrical energy accumulator. For this, energy conversion units such as compressors, Peltier

elements, piezo elements, and the like that are attached to the rotor shaft can be used. The auxiliary units, which were used, for example, as air conditioning compressors, can also be provided for this.

[0079] Another beneficial variation of the invented method can include a feature that the first electrical machine additionally or alternatively to the drive unit, which can be a second electrical machine or an internal combustion engine, transmits torque to the first transmission input shaft for driving the motor vehicle and from there through a gear pair between the first transmission input shaft and transmission output shaft to at least one driving wheel. The pair of gears can either be selected based on the driving situation or the gear pair of the currently-engaged gear can be used.

[0080] The method in accordance with the invention furthermore provides a feature that the first transmission input shaft with the first electrical machine is decelerated during shifting processes for the synchronization of the gears, and thus the moment of inertia of the rotor of the electrical machine is reduced so that the synchronizing devices are not overloaded and possibly can even eliminated, wherein the deceleration of the first transmission input shaft can occur by briefly closing the clutch between the drive unit and the first transmission input shaft, while the flow of torque between the drive unit and the driving wheels takes place through the second transmission input shaft. The extent of deceleration of the transmission input shaft depends on the synchronizing rotational speed of the first transmission input shaft that must be set. Monitoring of the synchronizing rotational speeds can occur through appropriate rotational speed sensors that are attached to the

transmission input shaft such as a sensor that is already incorporated in the electrical machine for controlling it and/or on the transmission output shaft and/or on the driving wheels as wheel rotational speed sensors, wherein when mounting them to the transmission output shaft they are accordingly calculated while taking the gear ratios of the engaged gear between both shafts into consideration. Furthermore a feature may be included where with non time critical upshifting processes, i.e., during shift processes that occur toward overdrive with regard to their gear ratio, electric synchronizing procedures take place exclusively, while in the case of down-shift processes exclusively mechanical synchronizing procedures take place. This procedure includes among other things the advantage that the expenditure of electrical energy is minimized during down-shifting processes and that electrical energy can be gained when upshifting due to a delay of the transmission input shaft. Acceleration of the transmission input shaft to the synchronizing rotational speed when down-shifting can occur, for example, by briefly engaging the appropriate clutch. Of course, synchronization can occur both mechanically and electrically in the case of time-critical shifting processes.

[0081] The shift sequence from a gear with lower gear ratio to a gear with higher gear ratio for a transmission with an appropriate method takes place by engaging, for example, the gear with low gear ratio between the first transmission input shaft and the transmission output shaft, engaging the clutch between the drive unit and first transmission input shaft and thus transmitting torque from the drive unit through the clutch to the transmission input shaft, from there through the gear pair to the transmission output shaft and from there to the driving wheel. During this

time, the next gear is engaged on the second transmission input shaft with a disengaged clutch between the drive unit and second transmission input shaft, wherein synchronization of the second transmission input shaft can be supported by a slipping contact of the clutch between the drive unit and second transmission input shaft or – if the second electrical machine is arranged on this transmission input shaft – by accelerating or decelerating the electrical machine. Of course, upshifting into the next gear can occur in the same fashion, i.e., by first transmitting torque to the driving wheel through the second transmission input shaft, while engaging the next gear on the first transmission input shaft, and then engaging the clutch to the first transmission input shaft and disengaging the clutch to the second transmission input shaft. The shift sequence from a gear with higher gear ratio to a new gear with lower gear ratio takes place in a similar fashion, i.e., by engaging and synchronizing the next lower gear on the transmission input shaft that is not connected with the drive unit through the clutch, with the clutch then interrupting the flow of torque through the engaged gear and starting the new gear by engaging the clutch to the transmission input shaft with the new gear.

[0082] Another beneficial shift variation can include the shifting from a gear with higher gear ratio to a gear with lower gear ratio on one and the same transmission input shaft, i.e., a downshift on the same transmission input shaft, which can be performed beneficially with the following procedural steps:

[0083] adjustment of the drive unit to increased power, preferably full load;

[0084] slipping operation of the clutch in power distribution flow between a transmission input shaft on which the gears that are to be shifted are to be arranged and the drive shaft;

[0085] upon reaching the synchronizing rotational speed for a gear that is between the gears on the one transmission input shaft with regard to its gear ratio on the clutch between the drive shaft and the other transmission input shaft, this clutch is operated in a slipping manner and torque with regard to its gear ratio between the gears on the gear on the one transmission input shaft is directed to at least one driving wheel through the transmission output shaft;

[0086] the clutch between the drive shaft and the one transmission input shaft is engaged;

[0087] upon reaching the synchronizing rotational speed of the new gear that is to be engaged on the one transmission input shaft the shift process to this gear takes place.

[0088] It may prove beneficial when shifting from one gear to a new gear with lower gear ratio on the same transmission input shaft that is to be engaged to additionally use the electrical machine during the synchronizing process to the new gear that is to be engaged if the electrical machine is operatively connected with this transmission input shaft. For the synchronization of at least one new gear that is to be engaged, which preferably is the gear with the smallest gear ratio on the transmission input shaft with which the electrical machine is connected from a drive point of view, it may furthermore be beneficial to employ the electrical machine for decelerating the transmission input shaft that is connected with it while accelerating

the motor vehicle through the transmission input shaft without the electrical machine. The transmission input shaft is preferably decelerated substantially to the synchronizing rotational speed of the new gear that is to be engaged.

[0089] Based on the inventive idea, another beneficial method that can be applied for arrangements of transmissions allows an electrical machine to be connected with a transmission input shaft through a shift clutch, which at the same time can connect the gear with the largest gear ratio with the transmission output shaft. This shift clutch undergoes the following shift modes:

[0090] the idler of the gear pair of the gear is rotatably arranged on the transmission input shaft, and the electrical machine is uncoupled from the transmission input shaft;

[0091] the electrical machine is uncoupled from the transmission input shaft;

[0092] the electrical machine is coupled to the transmission input shaft, the idler can be rotated in relation to the transmission input shaft;

[0093] the idler is non-rotatably connected with the transmission input shaft, and the electrical machine is coupled with the transmission input shaft;

[0094] the electrical machine is connected with the idler, and the idler can be rotated in relation to the transmission input shaft.

[0095] Based on the idea of the invention, the method furthermore provides beneficial steps for the sole operation of the motor vehicle with the first electrical machine or its operation that supports the internal combustion engine and/or a second electrical machine in its place. The clutches between the drive shaft and the transmission input shafts can be disengaged, and in accordance with the driving

situation a selected gear pair that is connected between the transmission input shaft and the transmission output shaft can transmit torque from the electrical machine to at least one driving wheel. Additionally the method can provide for support by the first electrical machine to the drive unit for operating the motor vehicle so as to allow the first electrical machine to directly interact with the transmission input shaft in a power distribution flow from the drive shaft to the transmission output shaft through the transmission input shaft, which can be coupled with the first electrical machine, to disengage the clutch between the drive shaft and the transmission input shaft with the electrical machine with a power distribution flow through the transmission input shaft without the electrical machine, and to transmit the torque fed by the electrical machine to the transmission output shaft through a gear pair that has been selected in dependence of the driving situation. Furthermore, in the case of a transmission arrangement with an electrical machine that can be shifted between the transmission input shafts, it may be particularly beneficial to connect the electrical machine of the transmission input shaft that does not transmit any torques from the crankshaft to the transmission output shaft at that time, and to operate it through one of the gears that are arranged on this transmission input shaft at a gear ratio that is optimal for efficiency.

[0096] Based on another idea of the invention, a creeping movement of a vehicle with the inventive double clutch transmission, i.e., a slow forward motion of the motor vehicle from the stopped position, such as during congested or stop-and-go traffic or the like traffic situations, may be beneficial. The starting situation can be a vehicle with a selected driving mode with an engaged gear and brakes applied,

wherein the internal combustion engine is not in operation. In accordance with the idea of the invention, a differentiation can now be made as to whether the motor vehicle is supposed to be operated in the creeping mode or quickly accelerated. The starting process for the internal combustion engine can be initiated either in dependence on the releasing of the brakes and/or by indicating a driving request, e.g., by signaling a load requirement to the internal combustion engine, such as by the actuation of the accelerator pedal, such as the gas pedal. Compared to the exclusive evaluation of the accelerator pedal, such as the gas pedal, this can save time, and the internal combustion engine can be started sooner. Additionally, a creeping movement of the vehicle can be excluded and it can be accelerated immediately when releasing the brakes quickly and actuating the gas pedal quickly, while during a slow release of the brake pedal a creeping process can be initiated. Depending on the behavior of the driver, a differentiation can be made among the following incidents with the respective subsequent procedural steps:

[0097] release the brake pedal, no actuation of the gas pedal after a stop phase:

[0098] the available torque of the electrical machine is transmitted through the clutch between the transmission input shaft with the electrical machine and the crankshaft;

[0099] between the other transmission input shaft and the transmission output shaft, torque that is sufficient for a creeping movement of the vehicle is transmitted to the transmission output shaft simultaneously through a gear with a

low gear ratio, i.e., the gear with the lowest gear ratio, in the case of a slipping clutch between this transmission input shaft and the crankshaft;

[00100] after starting the internal combustion engine, the internal combustion engine provides the creeping moment and the electrical machine is switched off.

[00101] The transmitted torque on the clutch between the transmission input shaft without the electrical machine and the crankshaft may be reduced down to zero if necessary in order to achieve a faster start of the internal combustion engine.

[00102] the driver actuates the accelerator pedal;

[00103] with released brakes and engaged clutch between the transmission input shaft with the electrical machine and with an engaged low gear between this transmission input shaft and the transmission output shaft, the creeping moment is generated through the electrical machine and directed to the driving wheels;

[00104] upon actuation of the gas pedal the engaged gear is deactivated;

[00105] the clutch between the crankshaft and transmission input shaft with the electrical machine is engaged;

[00106] a gear with low gear ratio between the transmission input shaft without the electrical machine and the transmission output shaft is engaged;

[00107] the internal combustion engine is started by the electrical machine;

[00108] the clutch between the crankshaft and the transmission input shaft with the electrical machine is disengaged after the start, the other clutch is engaged and the vehicle starts to move.

[00109] Of course, in accordance with the statements provided under a), here as well the clutch that needs to be engaged after the start can be adjusted to a defined creeping moment.

[00110] Advantageously, the clutch between the transmission input shaft with the electrical machine and the crankshaft is adjacent to the tactile point in order to shorten its engagement time.

[00111] Further beneficial embodiments, in particular for increasing shifting comfort and dynamics, e.g., for double and/or triple upshifting and/or down-shifting without a drop in tractive force, can include the integration of additional and/or the separation of existing transmission input shafts or secondary shafts with additional gear pairs.

[00112] Another advantageous shift sequence is provided by the invention in such a way that during a shifting process between a first gear, e.g., the gear II on a first transmission input shaft, and a second gear with a higher gear ratio than the first gear, e.g., the gear III on a second transmission input shaft, the clutch between the crankshaft and the first transmission input shaft transmits torque to the electrical machine that is operatively connected with the clutch until the crankshaft has achieved roughly the rotational speed that is required for a jerk-free operation of the second gear. This way, disadvantageous speed adjustments of the internal combustion engine can be avoided, for example, by reducing the load of the internal combustion engine by adjusting the ignition timing – connected with a shortened life of the catalytic converter due to an increased concentration of unburned hydrocarbons. The advantage of the electrical machine over this method consists

particularly of its clearly improved controllability and high dynamics, so that the acceleration gradients to the drive shaft can be kept low and an increased feeling of comfort arises when shifting. Furthermore the energy supply to the clutches can be reduced, which leads to an increased life and lower fuel consumption.

Brief Description of the Drawings

[00113] The invention is explained more in detail in Figures 1 through 35b. They show:

[00114] Figures 1 through 10, beneficial embodiments of a double clutch transmission with an internal combustion engine and an electrical machine in diagrammatic view as well as details,

[00115] Figure 11, a diagrammatic view of a double clutch transmission with two electrical machines,

[00116] Figures 12 and 13, graphs of the speed dependent power of the electrical machines,

[00117] Figures 14 through 18, torque curves of embodiments in accordance with the invention as a function of the shifting time between two gears,

[00118] Figure 19, a flow chart for completing a recouperation process,

[00119] Figure 20, another embodiment of a double-clutch transmission,

[00120] Figures 21 and 22, beneficial embodiments of a double-clutch transmission based on the idea of the invention, in diagrammatic view,

[00121] Figure 23, a diagrammatic view of an embodiment of an end actuating mechanism for the double-clutch transmission of Figure 2,

[00122] Figure 24a, a motor vehicle with an automatically actuated clutch and transmission,

[00123] Figure 24b, a motor vehicle with a branched drive train,

[00124] Figure 25, end output mechanisms with an end actuating mechanism,

[00125] Figure 26a, operation of an auxiliary actuating element,

[00126] Figure 26b, operation of an auxiliary actuating element,

[00127] Figure 26c, operation of an auxiliary actuating element,

[00128] Figure 26d, operation of an auxiliary actuating element,

[00129] Figure 27, a graph showing the selector shaft rotation angle and the clutch sleeve movement,

[00130] Figure 28a, an arrangement of a main actuating element and an auxiliary actuating element on a selector shaft,

[00131] Figure 28b, an arrangement of a main actuating element and an auxiliary actuating element on a selector shaft,

[00132] Figure 29a, an arrangement of a main actuating element and two particularly wide auxiliary actuating elements for actuating two end output mechanisms simultaneously,

[00133] Figure 29b, an arrangement of a main actuating element and two particularly wide auxiliary actuating elements for actuating two end output mechanisms simultaneously,

[00134] Figures 30a through 30e, embodiments of auxiliary actuating elements,

[00135] Figures 31a through 31 d, selector shaft position and H-shift pattern,

[00136] Figures 32a and 32b, selector shaft position and H-shift pattern with a wide auxiliary actuating element,

[00137] Figure 33a, an embodiment of the invention for application in a conventional manual transmission,

[00138] Figure 33b, a sleeve of an actuating element,

[00139] Figure 34a, an embodiment of the invention for application in an automated transmission,

[00140] Figure 34b, a side element,

[00141] Figure 34c, a bushing-shaped element,

[00142] Figure 35a, an embodiment of the invention for application in a double-clutch transmission, and

[00143] Figure 35b, a side element.

Description of the Preferred Embodiments

[00144] Figures 1 through 10 show in diagrammatic view several embodiments of double-clutch transmissions 1a through 1m, which should not be considered restrictive. The double-clutch transmissions 1a through 1m each contain two transmission input shafts 2a, 2b, as well as at least one output shaft 3, and/or 3a, 3b in Figure 2a, which is connected with at least one driving wheel, preferably two or four driving wheels, through a differential, a torque splitting mechanism such as a viscous-type clutch, a torque splitting transmission, and/or the like from a drive point of view, and thus transmits the driving torque to at least one driving wheel for the purpose of moving the vehicle, wherein a thrust torque that

is supplied by the wheels for recouperation purposes can also be supplied to the transmission in a reverse torque direction. Between the crankshaft 4 that is driven by an internal combustion engine and the transmission input shafts 2a, 2b a friction clutch 5, 6, respectively, is incorporated, which provides for the appropriate transmission input shaft 2a, 2b to be uncoupled from the crankshaft 4. In the course of the torque transmitted between the crankshaft 4 and the clutches 5, 6, respectively, a damping device can be provided for damping torsional vibrations and/or axial or wobble vibrations, for example, a two-mass flywheel 7a that is arranged between two crankshaft branches 4, 4a, or a torsional vibration damping device 7b in a clutch disk. Of course, the two-mass flywheel – as is basically known – can be integrated in at least one, preferably both clutches 5, 6, wherein in a preferred embodiment a two-mass flywheel with double clutch, as in Figures 5, 8, 9, 10 as a two-mass flywheel with double clutch 7c, can be particularly beneficial. The clutches 5, 6 are preferably formed as friction clutches with a pressing plate and a pressure plate that is connected with the pressing plate in an axially displaceable, non-rotatable manner, respectively. In special applications, wet clutches, e.g., in a multiple disk design, or similar to the converter bridging clutches of torque converters, which can be integrated into the transmission, can be advantageous. Of course, all advantages with regard to the design of converter bridging clutches, such as, e.g., profiled friction linings, piston controls for the piston controlling the converter bridging clutch, friction lining cooling and the like can be beneficial. When employing friction clutches friction linings are provided axially between the pressure plate and pressing plate, which are non-rotatably fastened with a clutch disk that is

non-rotatably connected with the respective transmission input shaft 2a, 2b. The frictional engagement between the pressure plate and pressing plate on one hand, and the friction linings on the other hand, is preferably accomplished through an axially displaceable energy accumulator that applies axial tension onto the pressing plate and pressure plate, e.g., a disk spring, which is actuated preferably axially through a disengaging device, wherein the prestress between the pressing plate, the friction linings and the pressure plate, and thus frictional engagement between the crankshaft 4 and transmission input shaft 2a, 2b, is eliminated with a disengaged clutch. Of course, when employing a double clutch 7c, a pressing plate can be provided for both clutches 5, 6, and a disengaging device can actuate both clutches and slipping clutch modes between an engaged and disengaged clutch can be adjusted with reduced transmittable torque. With regard to a double clutch 7c that can be utilized, a self-adjusting clutch can be provided, which is explained in detail and described in DE 100 17 815.4, which is hereby included in the present application with its entire content.

[00145] At least one disengaging device can be actuated automatically through an actuator. The actuator can be operated electrically, hydraulically, pneumatically, or in a combination of these methods, wherein, for example, an electric actuator can supply a master cylinder, which transmits the actuation impulse through a hydraulic line to a slave cylinder, which displaces the disk spring axially by an intermediate release bearing. Additionally, an electrical actuator can be arranged as a rotatable drive directly on the transmission input shaft 2a, 2b for an

axial drive, wherein one or two axial drives, which can telescopically arranged, can actuate the clutches 5,6.

[00146] Between the transmission input shafts 2a, 2b and the transmission output shaft 3, the gear steps I, II, III, IV, V, VI, R are provided for forming a transmission 1a through 1k with six forward and one reverse gear in this case, wherein these are arranged on the transmission input shafts 2a, 2b in an alternating fashion with regard to their gear ratios. The reverse gear R is arranged on the transmission input shaft 2b in the depicted examples 1a through 1k. This results in a shift process of the gears in such a way that, for example, a gear I can be engaged on the transmission input shaft 2b, the clutch 6 can be engaged, and the next gear II can already be engaged with a disengaged clutch 5 while driving the vehicle via the transmission input shaft 2b and the transmission output shaft(s) 3, 3a, 3b with the gear I, and only the clutch 5 is engaged and the clutch 6 is disengaged at the moment of shifting without tractive force interruption. In order to increase driving comfort, for example, the clutches 5, 6 can be operated in an overlapping manner, i.e., that in a certain operating range both clutches 5, 6 transmit torque from the internal combustion engine to the transmission output shaft 3 in a slipping operating mode. Further beneficial transmission designs of a double-clutch transmission are described in DE 100 25 878.6, which is hereby included in this application in its entire content.

[00147] Based on the inventive idea, an electrical machine 10 is operatively connected with the transmission input shaft 2a or is arranged around it in a connectable manner. In the illustrated examples, the rotor 9 with the rotor shaft 9a

is arranged radially within the stator 11, whose housing is firmly connected with the transmission housing or with another stationary component.

[00148] The examples of the double clutch transmissions 1a through 1k of Figures 1 through 10 differ from each other basically in the different arrangements of the electrical machine 10, the arrangement of the gears I – VI, R, and in part different operating modes resulting from these arrangements. The different double-clutch transmissions 1a through 1k are explained in more detail below.

[00149] Figure 1 shows a double clutch transmission 1a with a two-mass flywheel 7a that is provided between the crankshaft branches 4, 4a. In the diagrammatic view, the crankshaft branch 4a – as shown here – splits through a connection forming a positive lock, such as a gear connection, with a gear 4b that is coaxial to the crankshaft branch 4a and two gears 4c, 4d meshing with it, which are each arranged coaxially on an input branch 4e, 4f for the clutches 5, 6 of the transmission input shafts 2a, 2b, wherein between the gears 4b, 4c or 4b, 4d the gear ratio i=1 or a gear ratio differing from i=1 can be adjusted and wherein also the gear ratios i between the gears 4b, 4c and the gears 4b, 4d can be different and thus have a different gear ratio (multiplication or reduction) between the transmission input shafts 2a, 2b. Of course, the arrangement of the shafts 2a, 2b, 3 in one plane shown here does not necessarily prove beneficial for all transmissions of this type, but rather the shafts may require a small space in a spatial arrangement relative to each other. Furthermore the transmission input shafts 2a, 2b can be designed as shafts that are arranged around each other, wherein one transmission input shaft 2a, 2b is designed as a hollow shaft in which the other one

is guided. The two clutches 5, 6 separate the transmission input shafts 2a, 2b from the crankshaft 4 and thus prevent the torque connection to and from the internal combustion engine in the disengaged state.

[00150] On the transmission input shaft 2b the idlers 12, 13, 14, 15 are arranged in a rotatable manner relative to the clutch 6 starting with the smallest gear ratio (gear I) in an ascending gear ratio manner for the purpose of forming the gears I, III, V, R, and are positioned through the shift sleeves or sliding sleeves 16, 17, which engage two gears I, III or V, R, respectively, by connecting one of the idlers 12, 13 or 14, 15, respectively, in the conventional manner with the transmission input shaft 2b, or are in a neutral position in which no gear is engaged. The idlers 12, 13, 14, 15 mesh with one of the fixed gears 18, 19, 20, 21, respectively, which are non-rotatably arranged on the output shaft 3, for the purpose of forming the gear ratios of the gears I, III, V, R, wherein for the purpose of forming the reverse gear R a reversing wheel 22 meshes with both the fixed gear 21 and the idler 15. The shift sleeves 16, 17 are equipped with a synchronizing device 23, 24, 25, 26 such as a synchronizing ring, which can have a conventional design, for the odd-numbered gears I, III, V that are arranged on the transmission input shaft 2b, with the reverse gear R, while the gear I has the smallest gear ratio and can be designated as underdrive.

[00151] On the end opposite the clutch 5 the electrical machine 10 is connected with the transmission input shaft 2a through its rotor shaft 9a through a positive lock in the circumferential direction, e.g., it is flanged, has axial teeth, or the like. The electrical machine 10 can be arranged outside the transmission housing,

wherein the rotor shaft 9a or the transmission input shaft 2a, which is guided outward, is sealed against the housing. Alternatively, the electrical machine 10 can be incorporated in the transmission housing, wherein it may prove beneficial to encapsulate it separately.

[00152] Additionally, the even gears II, IV, VI are arranged on the transmission input shaft 2a, wherein the gear II is arranged between gear I and III with regard to its gear ratio, gear IV between gear III and gear V, and gear VI as overdrive with the largest gear ratio. For the formation of the gears II, IV, VI the idlers 27, 28, 29 are rotatably arranged on the transmission input shaft 2a and can be non-rotatably connected with the transmission input shaft 2a through the shift sleeves 30, 31, wherein the shift sleeve 30 can selectively shift one of the two gears II or IV or can be in a neutral position in which neither of the two gears II, IV is engaged, and the shift sleeve 31 engages the gear VI with the highest gear ratio or is in a neutral position. The idlers 27, 28, 29 mesh with the same fixed gears 18, 19, 20 as the idlers 12, 13, 14 of the transmission input shaft 2b. The gears II, IV, VI can be synchronized in the same manner as the gears I, III, IV of the transmission input shaft 2b through synchronizing devices (not shown). Alternatively they can be eliminated, wherein a synchronization of the idlers 27, 28, 29, which are coupled to the speed of the transmission output shaft 3 through the fixed gears 18, 19, 20, occurs through the electrical machine 10, which drives or decelerates the transmission input shaft 2a accordingly to achieve the synchronizing rotational speed.

[00153] The shift sleeves 16, 17, 30, 31 are actuated through appropriate shift forks (not shown), which slide them axially along the transmission input shafts 2a, 2b.

[00154] Actuation of the shift forks occurs automatically through one or several actuators (also not shown), for example, through appropriate kinematically controlled electric motors and/or electrical, hydraulic and/or pneumatic valves. It may prove beneficial to use an actuator not for every sliding sleeve, but one actuator for the selection movement for the purpose of selecting a shift fork for a sliding sleeve 30, 31 or 16, 17, respectively, and another actuator for the shift movement of the selected shift fork and thus the shift sleeve, so that for four actuators, two selection actuators and two shift actuators, respectively, are used for the entire shifting process of the transmission 1a. Furthermore it may also prove beneficial to combine the two selection actuators and the two shift actuators into one actuator, respectively, wherein the inventive idea provides for a gear to become engaged on the one transmission input shaft 2a, 2b without disengaging an engaged gear on the other transmission input shaft 2b, 2a, which is also activated in the same shift and selection arrangement. Such an arrangement of actuators with appropriate kinematics is explained and described in more detail in DE 100 20 821.5, which is hereby included in the present application in its entire content. Another beneficial embodiment can be an axial drive with an electrical rotational drive, which is arranged around shift sleeves 16, 17, 30, 31 and thus does not require any additional devices for the transmission of motion such as rods and the like devices. Such an axial drive is described under Figure 23 of the German

application with reference number DE 100 15 205.8, which is hereby included into the present application in its entire content.

[00155] The function of the double-clutch transmission 1a is explained with examples based on typical operating modes such as cold starting and warm starting of the internal combustion engine, a typical upshifting process, a typical down-shifting process, up- and down-shifting process of gears arranged on a transmission input shaft 2a, 2b, the assistance function of the drive through the electrical machine 10, driving only with the electrical machine 10, and a generator function of the electrical machine 10, such as for recouperation.

[00156] A cold starting process, for example, with outside temperatures of below 0° C, can be performed with an impulse start in this embodiment. For this, when a forward driving motion is desired, initially both clutches 5, 6 can be disengaged and the sliding sleeves 17, 30, 31 in the neutral position. The shift sleeve 16 connects the idler 12 of the gear I that is non-rotatably connected with the transmission input shaft 2b with the first gear, i.e., gear I, is engaged. The electrical machine 10 is supplied with power and reaches the predetermined impulse rotational speed, e.g. 2000 to 6500 r.p.m. The impulse rotational speed can be adjusted in a variable or fixed manner in dependence on the engine characteristics, such as compression, displacement, number of cylinders, and/or the like, outside temperature, oil temperature, the rest period of the vehicle, the viscosity of the engine and/or transmission oil, and/or the like. The clutch 5 is engaged and the internal combustion engine is started. Immediately after the start, the clutch 6 is engaged and the vehicle starts to move. The electrical machine 10 then operates

as a generator, the electrical energy that is generated is passed on to an electrical energy storage device, such as an accumulator, a high current battery, a high power capacitor, and/or the like. Beneficial versions can include combinations of these with appropriate power electronics that are designed to store electrical energy over an extended period of time in a particularly effective manner and to receive high energy density with high efficiency and quickly in a short-term storage device, and to release it just as quickly again. For this, particular energy accumulation methods that use physical energy effects, such as charge distribution, the build-up of electro-magnetic fields, and the like are suited, while for the long-term storage of electrical energy especially electro-chemical material converters such as accumulators, batteries, or the like can be used advantageously, wherein an energy exchange can be controlled or precluded through appropriate, e.g., diode-like circuits for different charge conditions and voltages.

[00157] In a warm starting process in a warmed-up state, or at outside temperatures e.g., of above 0°C, acceleration of the electrical machine 10 to the impulse rotational speed can be eliminated and it can be started directly with an engaged clutch 5. This way, a considerably quicker start of the internal combustion engine can be accomplished. Of course, with a more powerful design of the electrical machine 10, e.g., depending on the size of the internal combustion engine at a torque of 100 Nm to 250 Nm, an impulse start can be eliminated as well, wherein a selection of the torque in dependence on the vehicle size and weight, between 80 and 200 Nm has proven particularly beneficial for the efficient use of

the electrical machine 10 as a starter generator with the utilization of recouperation, as well as the supportive and short-term sole operation of the vehicle.

[00158] As soon as the vehicle has started to move, e.g. in gear I, the clutch 5 is disengaged and the gear II is engaged through the shift sleeve 30. In order to activate the gears during an appropriate driving situation, for example, when reaching a certain rotational speed of the internal combustion engine, the clutch 5 is engaged and clutch 6 is disengaged. Similarly, the subsequent gears III to VI are engaged by engaging the subsequent gear with a disengaged clutch 5 or 6 and then activating it through a torque change from one transmission input shaft to the other by disengaging the one clutch and engaging the other clutch 5, 6. Down-shifting takes place in the reverse order. Selection of a subsequent gear can occur by evaluating the driving situation, such as speed, acceleration, direction of acceleration, the rotational speeds of the transmission input shafts, of the transmission output shaft, of the driving wheels, of the non-driven wheels, the cross-acceleration, the fuel consumption, the gas pedal position, the load of the vehicle, the trailing load, and/or the like parameters. For this it may prove beneficial to integrate a control device for the transmission 1a into an overall control device of the vehicle or to connect it with the vehicle and to evaluate the measured parameters and characteristic lines of additional vehicle components, such as sensor signals, characteristic lines of the internal combustion engine, of auxiliary units, the brake system, the fuel supply system, and/or the like.

[00159] In certain driving situations, it may prove beneficial to perform up- and down-shifting processes where one gear that is currently used and a desired gear

that is supposed to be engaged are arranged on the same transmission input shaft, e.g., transmission input shaft 2a, such as when shifting from gear II to gear IV, from gear IV to gear VI. For this purpose, the shift from gear II to gear IV on the transmission input shaft 2a is explained in more detail. After acceleration of the vehicle in gear II the clutch 5 is disengaged and in the meantime the clutch 6 with the engaged gear III is engaged, which allows the rotational speed of the internal combustion engine to be adjusted to the gear III and is thereby reduced. The transmission input shaft 2a, which in extreme cases can rotate at the nominal speed of the internal combustion engine, must be decelerated to the new synchronous speed for the gear IV. So as not to have to design possibly existing synchronous rings in an oversized manner or so as to avoid long synchronizing times in the synchronization provided for by the electrical machine, due to the poor efficiency at which the electrical machine 10 runs at these speeds, synchronization can take place by decelerating the transmission input shaft 2a by briefly engaging the clutch 5, wherein the braking torque of the transmission input shaft 2a is made available by the torque of the internal combustion engine. The course of the torque over time during this shifting process is shown in the diagram in Figure 14 with the courses of the torques 150 of the transmission output shaft 3, 151 of the internal combustion engine on the transmission input shaft 2a, 152 on the clutch 5, and 153 of the electrical machine 10. Between the times A and B, the shift sleeve 30 of the gear II is released without load at a disengaged clutch 5, in the area between the times B and C the torque 152 of the internal combustion engine is reduced to the extent that a braking torque 152 is built up on the clutch 5 and the transmission input shaft 2a

is decelerated to a specified rotational speed. For the purpose of further synchronization, a braking torque 153 is built up in the area between the times D and E through the electrical machine 10 until the synchronous rotational speed has been reached, so that during the time interval D-E the shift sleeve 30 for the gear IV can be engaged again without load. Afterward, the clutch 5 is engaged again, and the electrical machine 10 can in that case be operated as a generator again. The course of torque 150 of the transmission output shaft 3 remains constant throughout the shifting process due to the constant supply of the torque of the internal combustion engine through the engaged clutch 6 and the engaged gear III, so that the shifting process, which lasts preferably less than 1 second, or even preferably less than 0.7 second, occurs in a load-shifting manner. The corresponding rotational-speed-time behavior during this shifting process from gear II to gear IV is shown in the diagram in Figure 15 with the rotational speeds 160 of the transmission output shaft 3, 161 of the internal combustion engine, 162 of the idler 28 for the gear IV and 163 of the electrical machine 10 over the time shown. During the time interval A-B the electrical machine 10, and thus the transmission input shaft 2a, rotate at the speed at which the shift sleeve 30 was released and is decelerated due to the torque supplied by the internal combustion engine by engaging the clutch 5 in the time interval C-D. The idler 28 is driven by the increasing rotational speed 160 of the internal combustion engine, which drives the transmission output shaft 3, also with increasing gear ratios in accordance with the gear ratio between the fixed gear 19 and the idler 28 at reduced rotational speed 162, so that the decreasing rotational speed 163 of the electrical machine 10, which generates a braking torque

for the transmission input shaft 2a for this purpose, and the rotational speed 162 of the idler 28 approach each other in the area of the time D of the synchronous rotational speed, and subsequently in the time interval D-E the shift sleeve 30 can engage the gear IV.

[00160] When down-shifting from an engaged gear to a gear on the same transmission input shaft, e.g., the transmission input shaft 2a, i.e., from gear VI to gear IV or from gear IV to gear II, for example when the vehicle is driven at a low rotational speed of the internal combustion engine and the driver desires quick acceleration, e.g., through a kick-down actuation of the accelerator pedal, the driving torque is directed through the transmission input shaft 2b for supplying the tractive force. The procedure for this shifting mode will be explained in more detail with the example of a down-shift from gear IV to gear II. Based on the load requirement, the internal combustion engine is first accelerated to full load, and the clutch 5 is disengaged only briefly for a load-free disengagement of the shift sleeve 30, and then again partially engaged, i.e., operated in a slipping mode, so that only a portion of the torque that is made available by the internal combustion engine is directed into the clutch 5 and thus into the transmission input shaft 2a. The clutch 5 can be operated in such a way that only a specified torque is transmitted to the transmission input shaft 2a. At least one measured variable that can be used for controlling the clutch 5 can be the rotational speed of the crankshaft 4, the transmission input shafts 2a, 2b and/or the transmission output shaft 3. Due to the limited supply of torque, the internal combustion engine increases its speed, which allows it to reach the synchronous speed for the gear III on the transmission input

shaft 2a. Initially, the clutch 6 is partially engaged, i.e., operated in a slipping mode, and the gear III is engaged through the shift sleeve 16, while the clutch 5 is completely engaged, wherein the internal combustion engine accelerates the transmission input shaft 2a to the new synchronous rotational speed of the gear II during optimal support of the electrical machine 10. After this synchronous speed has been reached, the clutch 6 is disengaged completely, and the gear II is engaged through the shift sleeve 30.

[00161] Figure 16 shows the course of rotational speed 170 of the transmission output shaft 3, 171 of the idler 27 of the gear II, 172 of the idler 28 of the gear IV, and 173 of the internal combustion engine during a shifting process from gear IV to gear II over time. At nearly constant speed 170 of the transmission output shaft 3, the speed 173 of the internal combustion engine is adjusted to the speeds 171, 172 of the idlers 27, 28, which differ due to their gear ratios, wherein the internal combustion engine on one hand supplies torque to the transmission output shaft 3 through the clutch 6 and the gear III, and on the other hand accelerates the transmission input shaft 2a through frictional engagement of the clutch 5 until at the point 171a the synchronous rotational speed between the idler 27 and transmission input shaft 2a has been reached.

[00162] The appropriate course of torque in dependence on time during the shifting process is shown in Figure 17. The course of torque 183 of the internal combustion engine shows an increasing supply of torque into the transmission 1a up to the time Z, at which the clutch 5 is disengaged and the gear II is engaged at the synchronizing rotational speed. Afterward, the clutch 5 is engaged in an

overlapping shifting process, and the clutch 6 is disengaged. The course of the torque curve 181 shows the moment of inertia of the rotor 9 of the electrical machine 10. At the point 181a, it is converted through the supply of torque by the internal combustion engine with a slipping clutch 5, while the transmission input shaft 2a is accelerated with the rotor 9.

[00163] Acceleration of the transmission input shaft 2a takes place much more quickly than when accelerated through the electrical machine 10, which can additionally be supplied with power to accelerate the shaft. The course of torque 180 of the transmission output shaft 3 is basically constant and experiences a torque conversion through the shifting process. Figure 18 shows the course of torque 191 of the torque applied to the clutch 5 during the shifting process from gear IV to gear II as well as the appropriate course of torque 190 for the clutch 6. Before initiating the shifting process at the time 0, the vehicle is operated through the clutch 5 and engaged gear IV with low torque and the gear II is engaged through the shift sleeve 16. At the time Z=1 the clutch 5 is disengaged and clutch 6 is engaged, then the gear IV is disengaged. After Z=2 the vehicle is operated through the clutch 6 and gear III, and the transmission input shaft 2a including the rotor 9 is accelerated through the slipping clutch 5. Upon reaching the synchronizing rotational speed at Z=3, the clutch is disengaged and the gear II engaged. At the time Z=4, the clutch 6 is disengaged, and the clutch 5 is engaged.

[00164] Furthermore, it may prove beneficial when starting to move the vehicle in the gear I to not engage the gear II immediately, but instead keep the clutch 5 engaged and drive the electrical machine 10 through this clutch and the

transmission input shaft 2a as a generator for the purpose of generating electrical energy until the driver actuates the gas pedal. Since the acceleration process in the gear I is very brief, the synchronizing and shifting process should therefore be completed in a relatively short period of time, e.g., in less than 1 s, preferably in less than 0.5 s. For this, before starting to move the vehicle the transmission input shaft 2a is accelerated with an engaged clutch 5 through the internal combustion engine that is accelerated to full load, and after starting to move the vehicle in gear I the clutch 5 is disengaged immediately and the rotating transmission input shaft 2a is delayed to the synchronizing rotational speed of the gear II by the electrical machine 10 in the generator mode, and/or a possible existing synchronizing device. Of course, the vehicle does not always have to start moving in gear I, particularly in the case of heavy vehicles it may be prove beneficial to start these moving with the gear II and to use the gear I only for very steep ascending slopes or as a creeping gear. In this and other cases of special versions of double clutch transmissions, it may be advantageous to provide the electrical machine on the transmission input shaft with the gear with the smallest gear ratio, for example, in this transmission 1b' the electrical machine 10 is provided on the transmission input shaft 2b.

[00165] When operating the vehicle under traction in a pulling mode, the electrical machine 10 can be operated as a generator for generating electrical power, as already mentioned above. Furthermore, during a deceleration mode, the electrical machine 10 can recuperate, i.e., gain energy during generator operation from the kinetic energy of the vehicle, which is directed into the transmission 1a through the transmission output shaft 3. Both clutches 5, 6 can be disengaged,

wherein in dependence on the speed of the vehicle a suitable gear II, IV or VI can be engaged for optimal efficiency at nominal rotational speed of the electrical machine 10. Of course, it may prove beneficial not to uncouple the internal combustion engine in certain driving situations, particularly when the generation of electrical energy is not required, for example, with a fully charged accumulator. Furthermore, the internal combustion engine can additionally be connected, e.g., in a slipping mode, for controlling a recouperation torque, such as in the case of a slippery road and/or to achieve constant retardation in ascending or descending slopes. Furthermore, in a pulling mode of the internal combustion engine, the electrical machine 10 can be operated as a generator with optimal speeds near the efficiency optimum with a disengaged clutch 5 and a flow of torque via the transmission input shaft 2b through one of the gears II, IV, VI. Figure 12 shows the typical rotational speeds 201 of the gear II, rotational speed 202 with an engaged clutch 5 without an engaged gear, rotational speed 203 of the gear IV and the rotational speed 204 of the gear VI at a rotational speed of the internal combustion engine of about 1500 r.p.m. in gear III, wherein the gears II, IV, VI connect the electrical machine 10 with the transmission output shaft 3, respectively. The efficiency curve 210 of a typical electrical machine 10 in dependence on the rotational speed illustrates that in this example the gear II at the rotational speed 201 achieves the best efficiency. Figure 13 depicts the typical rotational speeds 201a (gear II), 202a (all gears in neutral position, clutch 5 engaged), 203a (gear IV), 204a (gear VI) at a rotational speed of the internal combustion engine of about 4000 r.p.m. with gear III as the driving gear, and it becomes clear that optimal efficiency

of the efficiency curve 210 of the electrical machine 10 is best approached with gear IV. Of course, at any gear I, III, V, in dependence on the rotational speed of the internal combustion engine, different gears II, IV, VI or a neutral position can achieve the best efficiency of the electrical machine 10 with an engaged clutch.

[00166] The example of a double clutch transmission 1b and an electrical machine 10 that is drivingly connected with the transmission input shaft 2a through the rotor shaft 9a, as shown in Figure 2, is basically similar to the transmission 1a of Figure 1 in its structure and function, but exhibits differences in the arrangement of the gears II, IV and VI on the transmission input shaft 2a, wherein they are arranged starting from the clutch 5 in the direction of the electrical machine 10 in a descending manner with regard to their gear ratios, i.e., gear IV is next to the clutch 5 and gear II is next to the electrical machine 10. Furthermore, separate fixed gears 18a, 20a are non-rotatably arranged on the transmission output shaft 3 for the gears I and VI, with these gears not being used by a gear on the other transmission input shaft 2a or 2b. The gears I and III, IV and VI, as well as V and R, are engaged through a shift sleeve 16a, 17a, 30a that is provided for two gears by non-rotatably connecting them with the respective transmission input shaft 2a, 2b. The arrangement of the fixed gear 18a' and the idler 27a for the purpose of forming the gear II takes place in the opposite manner, so that the fixed gear 18a is arranged on the transmission output shaft 2a and the idler 27a is arranged on the transmission output shaft 3 in an articulating manner and can be non-rotatably connected with this shaft through the sliding sleeve. Another special feature of the present transmission structure is the formation of the gear V through two idlers 14a, 20a',

wherein – as already indicated – the idler 14a is arranged on the transmission input shaft 2b and can be non-rotatably connected with it through the sliding sleeve 17a, and the idler 20a' is arranged on the transmission output shaft 3 and can be non-rotatably connected with it through the sliding sleeve 8. The sliding sleeve 8 hereby performs another function; in another shifting mode it non-rotatably connects the idlers 20a', 27a of the axially neighboring gears II, V with each other, wherein their rotational ability on the transmission output shaft 3 remains. This way the transmission input shaft 2a is connected with the transmission input shaft 2b from a drive point of view by forming a gear ratio that results from the quotient of the gear ratios of the gears II, V. Of course, the connection of other gears may also prove beneficial for attaining different gear ratios. In this example, the series connection of the gears II and V is preferably used to start the cold internal combustion engine directly through the electrical machine 10, particularly during low outside temperatures, with the necessary gear ratio without employing an impulse start. To achieve this, the clutch 5 is disengaged, and the clutch 6 is engaged. Through the shift sleeve 8, a connection between the transmission input shaft 2a and the transmission input shaft 2b is created through the gears II and V and the electrical machine 10 is supplied with power. The gear ratio condition i resulting from this power path through the division of the gear ratios of the gears II and V generally amounts to between i=2.5 and i=4, so that the electrical machine 10 can start the internal combustion engine with a smaller maximum torque than would be the case directly through the clutch 5. The remaining operating modes and functions are basically identical to the transmission 1a of Figure 1.

[00167] In order to reduce fuel consumption, the internal combustion engine is preferably uncoupled and shut off during pushing phases, e.g., by disengaging at least the clutch 5 and shifting the clutches and/or sliding sleeves 16, 17a of the transmission input shaft 2b into the neutral position and also disengaging the clutch 5. In certain driving situations, for example, during the warm-up phase of the internal combustion engine, shortly before a full load requirement or in general with an appropriate design of the vehicle, it may prove less beneficial to shut off the internal combustion engine, and it should therefore be pulled along in these vehicles during a recouperation process, wherein the pulling loss will have a negative effect on the energy that can be recuperated. It is additionally disadvantageous that the internal combustion engine must be operated near idle, e.g., 880 to 1400 r.p.m., in order to minimize the pulling loss, and that thus the electrical machine 10 must be operated at appropriate rotational speeds below optimal efficiency. In order to compensate for these disadvantages, the suggestion is made in example 1b shown in Figure 2 to pull the internal combustion engine along in the deceleration mode after an adjustable speed limit, e.g., 60 km/h, and to couple the electrical machine 10 to the transmission output shaft 3 through the gear II. This increases the rotational speed of the electrical machine 10 over that of the crankshaft by the quotient of the two gears II and IV, for example by a factor of 2.5, so that at a speed of about 60 km/h the internal combustion engine is pulled along at a speed of n=1700 r.p.m. and the electrical machine 10 can be operated at about 4200 r.p.m. Shifting hereby occurs through the sliding sleeve 8, which is designed so as to connect both 20a' of the gear V and the idler 27a of the gear II with the fixed gear

that is firmly connected with the transmission output shaft. The control unit is able to calculate the time at which a down-shift from gear VI into a lower gear is energetically favorable despite increased pulling torque and thus establish the speed limit. When the driving speed continues to decrease, another down-shift is preferably provided to effectively utilize the recouperation process. Based on the inventive idea, the down-shifting process however does not occur to the gear IV on the transmission input shaft 2a with the electrical machine 10, but into gear III, wherein the pulling torque of the internal combustion engine is maintained during the shifting process by a short-term slipping operating mode of the clutch 5 with an engaged gear II, so that the internal combustion engine does not stop during the shifting process and has to be re-started. This shifting technique achieves a considerably quicker process of putting the vehicle in motion after a full load requirement, with only a slightly worse energy balance.

[00168] Figure 2a shows a variation of a double clutch transmission 1b', which is particularly suited for a front crosswise installation in a vehicle. Transmission 1b' is similar to the double clutch transmission 1b of Figure 2 with a split transmission output shaft, which consists of two parallel branches 3a, 3b and the parallel transmission input shafts 2a, 2b in between, wherein the transmission input shaft 2b is a hollow shaft and arranged around the transmission input shaft 2a. Both branches 3a, 3b can be joined, for example, through a geared connection or in the differential. The arrangement of the gears I-VI, R is such that the fixed gears 18b and 19b are arranged on the hollow shaft 2b, which is shorter than the transmission input shaft 2a, wherein the fixed gear 18b – which is arranged next to the two-mass

flywheel 7c that contains the clutches 5, 6 as a double clutch - meshes with the idler 12b of the gear I on the branch 3a of the transmission output shaft and with the idler 21b of the reverse gear R on the branch 3b, through intermediate reversing gear 22b. The fixed gear 19b meshes with the idler 14b of the gear V on the branch 3b on the one hand and with the idler 13b of the gear III on the branch 3a on the other hand.

[00169] The fixed gears 18b', 20b, 20b' for the gears II, IV and VI are arranged on the transmission input shaft 2a, wherein the idlers 28b and 29c are arranged on the branch 3a for the purpose of forming the gears IV and VI and the idler 27b is arranged on the branch 3b, adjacent to the idler 14b, for the purpose of forming the gear II. On the end of the branch 3b opposite the output 3c another fixed gear 3d is provided, which forms a parking lock 30b through appropriate means preventing the rotation of the fixed gear 3d. The gears I and III, IV and VI, R and V are engaged through a sliding sleeve 24b, 25b or 29b, respectively, by forming a non-rotatable connection between the branches 3a and 3b on one hand, and the idlers 12b, 13b, 14b, 21b, 27b, 28b, 29c on the other hand. The sliding sleeve 8 engages the gear II and non-rotatably connects the idlers 14b of the gear V and idler 27b of the gear II to form a reduced power path described under Figure 2 along the dotted line 31b between the rotor shaft 9a of the rotor 9 of the electrical machine 10 and the crankshaft 4 in order to start the cold internal combustion engine, particularly at low outside temperatures. For this, all sliding sleeves with the exception of sliding sleeve 8, which connects with the two idlers 27b, 14b, are in a neutral position, the clutch 5 is disengaged and the clutch 6 is engaged.

[00170] In this example, the electrical machine 10 is not directly flanged to one of the transmission input shafts, but instead is connected with the idler 28b of the gear IV through a gear 32b. This offers the possibility of coupling the electrical machine 10 to the transmission input shaft 2a with the gear ratio of the gear IV and a specified gear ratio between the fixed gear 20b and the gear 32b, and thus couple it directly also to the internal combustion engine with an engaged clutch 5 through the crankshaft 4 and to start it directly or to absorb torque from it as a generator, wherein the idler 28b –except when driving in gear IV – is not non-rotatably connected with the branch 3a.

[00171] Figures 2b and 2c show different variations for engaging the gears II and V through the sliding sleeve 8a, 8b of the transmission 1b, 1b'. The idlers 22', 24' are rotatably arranged on a transmission output shaft or transmission input shaft and are positively connected by means of teeth 22a', 24a' with a fixed gear 34, which is non-rotatably arranged on shaft 33, through an axial displacement of the sliding sleeve 8a, 8b. Synchronizing member 35 located on a cone balances relative rotational speeds between the idlers 22', 24' and the fixed gear 34 – in a conventional manner – by forming a frictional engagement with the cone surfaces.

[00172] The task of the sliding sleeve 8a of Figure 2b consists of creating a synchronized positive lock between the fixed gear 34 and the idler 24' from the depicted neutral position through an axial displacement in the direction of the idler 24', wherein the outer teeth 24a' enter into a positive lock with the inner teeth 8a' and outer teeth 34' of the fixed gear 34 and form a positive lock with additional inner teeth 8a''' of the sliding sleeve 8a. Additional axial displacement resolves the

second task of non-rotatably connecting the two idlers 22', 24' together, wherein the teeth 8a" of the sliding sleeve 8a and the teeth 22a' of the idler 22' on one hand, and the teeth 8a' of the sliding sleeve 8a and the teeth 24a' of the idler 24' on the other hand, enter into a positive lock. The two idlers are only connected with each during stopped positions of the shaft 33 and the idlers 22', 24', so that the formation of such a positive lock does not require synchronization.

[00173] The design of the sliding sleeve 8a does not provide for connecting the idler 22' with the shaft 33 and engaging the associated gear. This possibility is offered by the arrangement in Figure 2c, where the sliding sleeve 8b connects an idler 22', 24', respectively, with the fixed gear 34, and thus with the shaft 33, in a non-rotatable manner through an axial displacement from the depicted rest position. The connection of both idlers in this example takes place with a safety catch 36, which can be swung into the teeth 22a', 24' radially from the outside to form a positive lock, and can be moved mechanically, electrically, hydraulically, or pneumatically between two end positions. Additional beneficial embodiments in Figure 2b, particularly for shortening the space axially and reducing the axial expansion of the sliding sleeve 8a, can incorporate a feature that one of the idlers, e.g., idler 22', is rotatably located on the other 24', wherein the fixed gear 34 is arranged on an axial end of the idler 24', the idler 22' is seated axially between the teeth 24a' and the gear teeth 24b', and the teeth 22a' are arranged on the side of the idler 22' facing the fixed gear 34.

[00174] Figure 2d shows a particularly preferred embodiment of a double clutch transmission 1m as a so-called in-line transmission, particularly for front

longitudinal installation of the internal combustion engine with rear drive. The transmission input shaft 2b forms a hollow shaft around the transmission input shaft 2a, both transmission input shafts are arranged parallel to the crankshaft 4 and to the transmission output shaft 3, which is arranged coaxially and in an axial extension to the crankshaft. The transmission of power from the crankshaft 4 through the clutches 5, 6, which are associated with a transmission input shaft 2a, 2b, respectively, and have the design of a double clutch, takes place via the crankshaft branches 2104a, 2104a', which are connected with the clutches 5, 6, through a power transmission, e.g., a geared connection, which is formed by the gears 2104b, 2104b' of the crankshaft branches 2104a, 2104a' and the gears 2104c, 2104d, which are non-rotatably arranged on the transmission input shafts 2a, 2b. Formation of the gear ratios of the individual gears I, II, III, IV, V, VI, R takes place as in the examples in Figures 1, 2, 2a through an idler and a fixed gear, respectively, which can be arranged on one of the transmission input shafts 2a, 2b or on the transmission output shaft 3, respectively. Shifting of the gears takes place through shift devices with possibly provided synchronizing devices, wherein the gears I, III can be associated with the transmission input shaft 2b and the gears II, IV, VI, R to the transmission input shaft 2a. Features of the in-line transmission that differ from the previously-described examples are that one gear – in this case gear V – can be directly crossed over and can be engaged, e.g., by connecting the crankshaft branch 2104a and the transmission output shaft 3 through the clutch 2129, which has an end position to connect the two shaft 2104a, 3, and additionally

has a neutral position, in which the two shafts are separated, and an end position for engaging the gear I.

[00175] For spatial reasons, the electrical machine 10 is supported around the transmission output shaft 3, wherein the rotor 9 is rotatably supported in relation to the shaft and the stator 11 is firmly connected with the transmission housing and can be operated in accordance with the functions described in the figures described above. For shifting a suitable gear ratio between the crankshaft 4 and the rotor 9 particularly during a cold start of the internal combustion engine and through the electrical machine 10, a feature is provided for equipping the reverse gear R both with an input-side idler 2121b and an output- side idler 2118b, and furthermore for arranging this idler 2118b not directly on the transmission output shaft 3, but instead on a connection 2113c of the idler 2113b of the gear III, with the connection being rotatably supported around the transmission output shaft 3. In the depicted end position, the axially-displaceable triplex sleeve 2108 now connects the fixed gear 2119b, which is connected with the transmission output shaft 3, with the idler 2118b and engages the reverse gear R when the shift sleeve 2129 also connects the idler 2121b with the input-side fixed gear 2120b for the purpose of engaging the gears R and IV. When the sliding sleeve 2129b is in a neutral position or the gear IV is engaged, the idlers 2118b, 2121b can rotate freely.

[00176] The gear III is engaged by displacing the triplex sleeve 2108 axially in such a manner that it connects the idler 2113c with the fixed gear 2119b. The gear ratio for the cold start of the internal combustion engine is engaged by connecting the idler 2121b with the fixed gear 2120b through the sliding sleeve 2129b, wherein

the idler 2113c of the gear III is connected with the idler 2118b of the reverse gear R while being able to rotate freely around the transmission output shaft 3 by axially displacing the triplex sleeve 2108 in its end position facing the crankshaft 4. The power flow from the rotor 9 of the electrical machine 10 then takes place through the gears 132b, 128b on the transmission input shaft 2a and through the gear pairs of the gears R, III with an appropriate torque conversion for the gear ratios of these gears on the transmission input shaft 2b, and from there through the gears 104c, 104b with an engaged clutch 5 and a disengaged clutch 6 to the crankshaft 4. Of course, this way an impulse start is also possible by keeping the clutch 5 disengaged until the electrical machine 10 has been accelerated up to reaching the appropriate impulse rotational speed and then engaging it.

[00177] Figure 3 shows an example of a double-clutch transmission 1c that corresponds substantially to the double-clutch transmission 1a of Figure 1. The main difference is the arrangement of the electrical machine 10, which allows it to be uncoupled from the transmission input shaft 2a. This preferably occurs through the sliding sleeve of the gear, whose sliding sleeve in a six- or four-gear transmission does not engage two idlers of two gears. In the example shown, this is gear VI. The sliding sleeve 31c is arranged axially between the idlers 28, 29 and has four possible shifting positions. The first position is the neutral position shown, wherein neither gear VI is engaged nor is the electrical machine 10 coupled to the transmission input shaft 2a. The second position in the case of an axial displacement of the sliding sleeve 31c in the direction of the idler 29 connects the transmission input shaft 2a with the rotor shaft 9a of the electrical machine 10. The

third position with further axial displacement of the sliding sleeve 31c, connects the idler 29, the rotor shaft 9a and the transmission input shaft 2a with each other, e.g., when the vehicle is operated in gear VI and during generator operation of the electrical machine 10 or when the vehicle is operated with one of the gears I, III, V and a disengaged clutch 5. The fourth position, at the end position of the sliding sleeve 31c, connects the electrical machine 10 only with the gear VI, for example, or during a recouperation process with gear VI, or during operation of the vehicle with one of the gears I, III, V. The version that can be uncoupled from the transmission input shaft 2a is particularly beneficial with electrical machines 10, which are inexpensive and have therefore limited power. Such electrical machines cannot support time-critical synchronizing processes sufficiently with their own power supply, and are uncoupled during these shifting processes pursuant to the inventive idea.

[00178] Figure 4 shows another example of a double-clutch transmission 1d with electrical machine 10, which is similar to the embodiments of Figures 1a, 1c of Figure 1 and 4, wherein the main difference to the double clutch transmission 1a consists of the fact that the electrical machine 10 can be uncoupled from the transmission input shaft 2a and coupled to the transmission input shaft 2b. Coupling occurs through an actuator 40, which optionally connects one of the two transmission input shafts 2a, 2b with the rotor shaft 9a of the electrical machine 10 through kinematics, shown in diagrammatic view here with the switches 41, 42, or uncouples it from the two shafts 2a, 2b. The connection can take place through a belt, friction wheel, or chain drive, through a magnetic clutch, through a geared

connection, or the like. By switching it, the electrical machine 10 can optionally be connected directly with the internal combustion engine through one of the transmission input shafts 2a or 2b with an engaged clutch 5 or 6 and drive it, e.g. for starting, or it can be driven by it in the generator mode. Furthermore, the number of gears and thus the number of gear ratios for operating the electrical machine 10 – as shown in the examples in Figures 12 and 13 – is larger at optimal efficiency, so that through an appropriate switch of the electrical machine 10 from one transmission input shaft to the other, that shaft can be operated even more closely to its power optimum. Another advantage of this arrangement is the usage of every gear I through VI for recouperation, wherein the electrical machine 10 is connected with the gear that is to be used appropriate to the transmission input shafts 2a, 2b. Furthermore, before a double shift, i.e., a shift between two gears on the same transmission input shaft, the electrical machine 10 can be connected with the transmission input shaft that is not involved in the shifting process before starting the shifting process, and can compensate the tractive force interruption caused by disengaging the active and engaging the new gear at least partially by directing torque into the other transmission input shaft.

[00179] Figure 5 provides for a design of a double-clutch transmission 1e that is modified over the gear box 1a of Figure 1, according to which, e.g., for space or cost reasons, the two clutches 5, 6 are combined into a double clutch, wherein the double clutch in turn can be arranged on a two-mass flywheel 7c. With this structure, the splitting of the axis 2b and the thereby necessary gears can be relinquished, and in some embodiments better noise behavior can be achieved. In

this example, the transmission input shaft 2b is arranged basically coaxially to the crankshaft 4, the second transmission input shaft 2a as well as the transmission output shaft 3 are arranged parallel to it, wherein the transmission output shaft 3 is arranged spatially between the two transmission input shafts 2a, 2b. Transmission of the torque of the hollow shaft stub 2a', which is connected with the output part of the clutch 5 and arranged around the transmission input shaft 2a, to the transmission input shaft 2a occurs through a positive lock or frictional engagement, for example, through a fixed gear 4b', which is attached to the hollow shaft stub 2a' and meshes with a reversing gear 4c', which in turn meshes with a fixed gear 4e' that is non-rotatably connected to the transmission input shaft 2a. The functioning and shifting processes are basically identical to the gear box 1a of Figure 1.

[00180] Figure 6 shows the transmission 1a as another example 1f of double-clutch transmissions, where at least one auxiliary unit 60 is drivingly coupled with the electrical machine 10. Such versions offer the benefit that safety-relevant auxiliary units and/or units to increase comfort, such as air conditioning compressors, power steering pumps, brake force boosters and/or vacuum pumps, and/or the like can be coupled to the electrical machine 10 and thus continue to be supplied even with a stopped internal combustion engine, e.g., in standstill mode or during recouperation. So as not to increase the synchronizing times by the increased moments of inertia of the auxiliary units 60, it may prove beneficial to provide a disconnect-type clutch 61, which can be a shift, magnetic, or friction clutch, and is controlled electrically and/or by an actuator that is controlled in dependence on the appropriate shift conditions, between the electrical machine 10

and at least one auxiliary unit 60. Of course, the at least one auxiliary unit 60 can also be uncoupled for other beneficial reasons, such as saving fuel, e.g., an air conditioning compressor can be uncoupled when the air conditioner is not in operation. Furthermore, the auxiliary unit can be coupled to the electrical machine or uncoupled from it upon reaching or exceeding a specified rotational speed in order that the auxiliary unit operates within a beneficial operating range, for example, through a clutch operating in dependence on the rotational speed of the auxiliary unit's shaft, e.g., a centrifugal clutch. In the case of several auxiliary units, such a clutch can be arranged on a shaft of at least one auxiliary unit so as to position the entire belt pulley plane in a central – for example, by arranging it on the rotor shaft 9a – or decentralized manner.

[00181] Figure 7 shows another variation of a double-clutch transmission 1g, which is similar to double clutch transmission 1f of Figure 6, where the auxiliary unit 60 is connected with the rotor shaft 9a through a belt drive 62. With regard to its gear ratio between the auxiliary unit 60 and electrical machine 10, the belt drive can have a fixed or variable design, for example, it can be a CVT transmission, wherein an automatic control can be provided, for example, for adjusting optimal efficiency of the auxiliary unit 60. Furthermore it may prove beneficial to integrate further auxiliary units into this belt drive 62, apart from an air conditioning compressor. Here as well a disconnect-type clutch can be provided between the electrical machine 10 and the belt drive 62, or between the shaft 60a and the belt drive 62. Furthermore, it may be beneficial to operate the electrical machine 10 in both directions of rotation so as to dispense with, for example, a reverse gear R, wherein

the vehicle in the electrical operating mode is driven with one of the forward gears I or II by reversing the direction of rotation of the electrical machine 10. Auxiliary units 60 that can be operated only in one direction of rotation can still be employed if they are uncoupled during the reversal of the direction of rotation pursuant to the inventive idea, and/or if a free wheel is provided in the power flow between the electrical machine and a belt pulley plane, or only between the appropriate auxiliary unit and the electrical machine.

[00182] Figure 8 shows an example of a double clutch transmission 1h, which is particularly beneficial for front longitudinal installation, with an electrical machine 10, wherein the transmission input shaft 2b has the design of a hollow shaft and is arranged coaxially around the transmission input shaft 2a and the transmission output shaft 3 and parallel to their axes. With regard to its rotor shaft 9a, the electrical machine 10 is arranged parallel to those axes and can be coupled directly with the transmission input shaft 2a through the sliding sleeve 31d with a switchable, positive-lock connection. Furthermore, when maintaining this positive lock to the transmission input shaft 2a, optionally gear II or gear IV can be engaged through sleeve 31d. Another optional shifting variation of the shift sleeve 31d can be a neutral position, where the electrical machine 10 is uncoupled from the transmission input shaft 2a and from the gears II and IV.

[00183] The gears I, III, V, R are arranged in that sequence on transmission input shaft 2b spaced from the clutches 5, 6, which are integrated into one double clutch and are shifted as has been explained more specifically in connection with Figure 1. The transmission input shaft 2a is designed longer axially than the

transmission input shaft 2b, and on the protruding part the gears II, IV and IV are arranged in that sequence. The functioning and mode of operation take place in accordance with the previously-described transmission 1a with the difference that the electrical machine 10 can be uncoupled from the transmission input shaft 2a and coupled with the gears II or IV. By connecting the electrical machine 10 with the transmission input shaft 2a, the r.p.m. range of the electrical machine 10 can be utilized with an engaged clutch 5 while taking the transmission ratio between the transmission input shaft 2a and the rotor shaft 9a into consideration. In the case of a connection of the electrical machine 10 through the gear II or IV and with a disengaged clutch 5, the r.p.m. range of the transmission output shaft 3 can be used while taking the gear ratios of the gears II or IV into consideration.

[00184] Figure 9 shows an example of a double-clutch transmission 1i that is similar to the transmission 1h of Figure 8, with the basic difference that the electrical machine 10 is arranged radially around the clutches 5, 6, wherein the rotor is connected directly with the transmission input shaft 2a, for example it is interlocked with a disk part. Furthermore, the transmission input shaft 2a is arranged around the transmission input shaft 2b as a hollow shaft, which leads to a modified arrangement of the gears I-VI, R. The gears II, IV, VI are arranged on the clutch side part of the transmission 1i and the gears I, III, V and R are arranged on the part of the transmission input shaft 2b that is axially extended compared to the transmission input shaft 2a in the area of the output side part of the transmission 1i. A connection of the electrical machine 10 with the transmission output shaft 3 from a drive point of view can occur through the gears II, IV or VI.

[00185] Figure 10 shows a transmission 1k that is very similar to the transmission 1h of Figure 8, which basically differs from it by an electrical machine 10 that is seated coaxially on the end of the transmission input shaft 2a that is opposite the clutch 5.

[00186] Figure 11 shows a diagrammatic view of a double-clutch transmission 101 with two electrical machines 110a, 110b, which each drive a transmission input shaft 102a, 102b, respectively. In the example shown, the two electrical machines 110a, 110b are arranged opposite to each other, and the associated transmission input shafts 102a, 102b are arranged basically coaxially to each other between the electrical machines 110a, 110b. The transmission output shaft 103 is arranged parallel to their axes and is equipped with at least one output 103a for driving at least driving wheel. It is also possible to drive two driving wheels when connecting a differential. Furthermore the transmission output shaft 103 can be equipped with another output 103b, which basically corresponds to the output 103a, for which another driving wheel or appropriately a pair of driving wheels can be provided for the formation of a four-wheel drive with equal or differing torque supply to both axles, for example, 10% to 50% of the torque on the front axle and accordingly 50% to 90% on the rear axle. In this case, it may prove particularly beneficial to split the output shaft 103 into two drive lines 103c and 103d in order to be able to drive the two driving wheels either on one axle without differential gear or to achieve the above-mentioned distribution onto two axles, and to provide between them for means 120 that transmit different torques from a drive point of view, for example, a clutch, a torque splitting mechanism, or the like, so that optionally one of the two

outputs 103 can be operated by one electrical machine 110a, 110, respectively, with an open torque splitting mechanism 120, or one of the two or both electrical machines 110a, 110b drive the one output 103a, and optionally additionally a second output 103b with a transmitting-torque-splitting mechanism 120. The torque-splitting mechanism 120 can also be operated in a slipping mode and can be a shift clutch, a magnetic clutch, a friction clutch, or a viscous-type clutch. The torque-splitting mechanism 120 can furthermore be shifted automatically, for example, in dependence on the different wheel rotational speeds, e.g., during slipping operation, and designed to be closed, or it can be operated by an actuator 130, which is actuated in dependence on at least one operating parameter, e.g., slippage of the driving wheels, overheating of the wheels, ascending or descending slope, the efficiency of the electrical machines 110a, 110b, load, and the like. Another beneficial design of the transmission 101 can provide a feature that the two transmission output shafts 103a, 103b each drive a driving wheel, which eliminates a differential.

[00187] The gear box 101 has four gears I – IV in the example shown, wherein the gears I and III are engaged by connecting the idlers 112, 113 through the sliding sleeve 129, and the gears II and IV by connecting the idlers 114, 115 through the sliding sleeve 130, wherein they, in turn, are axially displaced through the actuator 130. In the example shown, a single actuator 130 is provided with an actuating device 131, with which both sliding sleeves 129 and 130, a parking lock 132, which can catch a fixed gear 126 of the transmission output shaft 103, as well as possibly the torque-splitting mechanism 120, are actuated. Of course, several actuators can

be used as well. The idlers 112, 113, 114, 115 mesh with an appropriate fixed gear 116, 117, 118, 119, respectively, that is arranged on the transmission output shaft 102 or on the branches 103c, 103d, wherein embodiments can also be beneficial in which at least one idler on the transmission output shaft 102 and the corresponding fixed gear are arranged accordingly on one of the transmission input shafts 102, 102b.

[00188] Additional embodiments can include beneficial transmission structures where one or several of the shafts 102a, 102b, 103, 103c, 103d are designed as hollow shafts and can be arranged coaxially to each other. This arrangement may prove particularly beneficial for the transmission output shafts 103c, 103d and/or the transmission input shafts 102a, 102b, wherein in this case the electrical machines 110a, 110b – in arrangements corresponding with the one shown in Figure 11 – can be arranged opposite from each other or in a modular unit, for example radially above each other. The electrical machines 110a, 110b can have the same design, with equivalent or different power. The gears I and II and/or the gears III and IV can have the same gear ratios for the purpose of forming two symmetrical transmission halves or can have different gear ratios for the purpose of forming a four-step transmission 101. Of course, the gear steps I through IV shown can be represented, for example, by two planetary gear sets that can be shifted through brake bands or shift clutches, wherein lower noise transmissions can be manufactured, and the statements that were made about gear ratios, shifting strategies, and arrangements can largely be transferred to this example.

[00189] The example of the transmission 101 that is shown can perform the following functions, which should not interpreted as being limited:

[00190] one driven axle

[00191] the transmission output shaft 103 is one piece, the output 103b and the clutch 120 are eliminated. The output 103a drives one driving wheel or two driving wheels through a differential.

[00192] the transmission output shaft 103 is divided into two branches 103a, 103b, the electrical machines 110a, 110b drive one driving wheel, respectively. The path balance in curves is adjusted by the electrical machines 110a, 110b, and a differential has been eliminated. The torque-splitting mechanism 120, for example, a torque-splitting transmission, a shifting, friction, magnetic or viscous-type clutch, can serve optionally as a limited-slip differential between the two outputs 103a, 103b.

[00193] two driven axles

[00194] the transmission output shaft 103 is one piece, the torque-splitting mechanism 120 is eliminated, both outputs have one driving wheel, respectively, or two driving wheels that are coupled through a differential, and one or both electrical machines 110a, 110b drive the outputs 103a, 103b as a permanent all-wheel drive.

[00195] the transmission output shaft 103 is a two-piece unit with branches 103c, 103d that operate independently from each other, one of the two electrical machines 110a, 110b drives an output 103a, 103b, respectively, which drives one driving wheel or two driving wheels, which are coupled through a differential. The two outputs 103a, 103b can have different output values. For this, the electrical

machines 110a, 110b can have different power levels or can be operated at different power levels. The gear ratios of the gears I through IV can be adjusted to the different output values.

[00196] The transmission output shaft 103 is a two-piece unit with branches 103, 103d that can be coupled with each other through a friction clutch 120, a torque-splitting mechanism, such as a torque-splitting transmission, a viscous-type clutch, a magnetic clutch, and the like. One or both electrical machines 110, 110b act on both outputs 103a, 103b in accordance with the torque-splitting mechanism. With a disengaged clutch 120 or torque-splitting mechanism, the appropriate electrical machine 110a, 110b can drive an output 103a, 103b as described above.

[00197] Operation of a vehicle with the transmission 101 occurs through the electrical machines 110a, 110b, which can feed torque to the transmission output shaft 103 simultaneously or individually and thus can drive the driving wheels. It may prove particularly beneficial to operate only one electrical machine 110a, 110b in the case of low partial load, preferably with the lower rotational speed when taking the adjustable gear ratios of the gears I – IV into consideration. The other electrical machine 110b, 110a can be brought along, or in the case of a four-wheel drive it can be uncoupled through the torque-splitting mechanism 120, wherein at the same time the appropriate driving wheels are uncoupled through the shift clutches 129, 130. Synchronization of the gears I – IV with the rotational speeds of the respective transmission input shafts 102a, 102b takes place by controlling the electrical machines 110a, 110b. When shifting from gear I to gear III the electrical machine 110a takes over the full drive torque, and when shifting from gear II to gear

IV it is the electrical machine 110b. It may prove beneficial to design the electrical machines 110a, 110a in such a way, e.g., with a power output from 15 to 50 kW, so that they can be overloaded briefly, for example, with 40 to 300% of their nominal power.

[00198] The electrical machines 110a, 110b are provided for motor and generator operation so that a driving mode and a recouperation mode are possible; additionally, generator operation can also be used for decelerating the transmission input shafts 102a, 102b when synchronizing the shifting process, and the released energy can possibly be fed to the other electrical machine directly, without intermediate storage in an electrical accumulator. In a beneficial version, such a drive can be employed both as support in a motor vehicle with an internal combustion engine, and also in a vehicle that is only operated with electrical energy, e.g., from a fuel cell.

[00199] Figure 19 shows in a flow chart an example of the course of the transition of a double clutch transmission – here for the example 1a of Figure 1 – during the transition from a recouperation phase into driving operation with a switched off, i.e., stopped, internal combustion engine, wherein beneficially a time delay, e.g., with a duration of 0.2 to 3 seconds, can be initiated before starting the internal combustion engine. For the purpose of the transition between recouperation and starting of the internal combustion engine, a subroutine 250 is started in a control unit for controlling the transmission 1a in a starting field 251, with v>0 applying for the speed v of the vehicle, a signal S indicating actuation of the gas pedal, and n(KW)=0 (internal combustion engine off) applying for the rotational

speed n(KW) of the crankshaft 4 (Figure 1), wherein two parallel subroutines 252, 253 are set in motion. The subroutine 253 sets the torque M(EM) of the electrical machine 10 in program step 254 to zero and controls the neutral position NEUTRAL of the sliding sleeve 30 or 31 of the engaged gear G(E), for example gear II, IV, or VI, i.e., the idler 27, 28, or 29 of the gear G(E) can be rotated in relation to the transmission input shaft 2a. In the subsequent program step 255, the clutch K2 (reference numeral 5, Figure 1) is engaged, and internal combustion BKM is started by supplying power to the electrical machine 10 through an impulse start.

[00200] Parallel to this, in the sub-routine 252 the new gear G(Z) that is to be engaged is determined from at least one operating value, for example, the vehicle speed, the position of the gas pedal, the air resistance, the previously-engaged gear, the rotational speed of the driving wheels and/or the non-driven wheels, the transmission input shafts 2a, 2b, the crankshaft 4 or a combination thereof in program step 256, and a power requirement 257 is issued in accordance with the new gear G(Z) that is to be engaged, and is evaluated in program step 258 for controlling the internal combustion engine BKM, wherein it is accelerated accordingly. Parallel to this, an examination takes place in the branches 259, 260 of the two routines 252, 254 as to whether the new gear G(Z) that is to be engaged is on the transmission input shaft GEW1 (2b, Figure 1). When the gear G(Z) is on the transmission input shaft 2b, G(Z) is engaged in step 261 and the two subroutines 252, 253 are combined in step 262, and when the crankshaft of the internal combustion engine reaches a higher speed n(KW) than the speed n(GEW1)

the clutch K1 (6 in Figure 1) is engaged in step 258 and the subroutine is terminated in step 263.

[00201] When the target gear G(Z) is not on the transmission input shaft GEW1, the clutch K2 is disengaged in the subroutine 253 in step 264, and the transmission input shaft GEW2 is synchronized with the new gear G(Z) that is to be engaged in step 265 through the electrical machine 10 and is engaged upon reaching the synchronizing speed. Parallel to this, the next higher gear G(Z+1) on the transmission input shaft GEW1 is engaged in the subroutine 252 in step 266, and subsequently the clutch K1 is operated in a slipping mode in step 267 until the synchronizing rotational speed between the transmission input shaft GEW2 and the new gear G(Z) that is to be engaged has been reached. During this time, drive torque is transmitted to the transmission output shaft 3 (Figure 1) through the gear G(Z+1). Upon reaching the synchronizing speed for the gear G(Z), the subroutines 252 and 251 are combined in step 268 and the clutch K1 is disengaged and the clutch K2 is engaged, and they are continued to the end of the program 263.

[00202] Deviating from the course of the shifting program 250, it may prove beneficial to engage a target gear already during recouperation, in dependence on the vehicle speed. It may prove particularly useful to engage the target gear in relation to an expected full load requirement.

[00203] Figure 20 shows an example of a double clutch transmission where a friction clutch between the crankshaft and one of the two transmission input shafts has been replaced by two clutches. In additional embodiments, both friction clutches can be replaced by clutches. Furthermore, in accordance with the

previously-described examples in Figures 1 through 10, double clutch transmissions with shift clutches in the power flow between the crankshaft and transmission input shafts can have an appropriate design, for example, with a split output shaft, with hollow shafts, as an in-line transmission, fixed gears and idlers with the appropriate shifting devices that can be actuated manually or automatically, can be integrally arranged on the transmission input shafts or the transmission output shaft, or in a mixed fashion on the transmission input shafts and the transmission output shaft.

[00204] The embodiment of this group of double clutch transmissions shown in Figure 20, where one or both friction clutches have been replaced by shift clutches, is a double-clutch transmission 1201, which is similar to the double clutch transmission 1b of Figure 2, wherein the function of the clutch 5 (Figure 2) is assumed by the shift clutch 1205a. An additional shift clutch 1205b connects the stator 1211 of the electrical machine 1210 optionally with the transmission housing 1250 – which is only schematically shown in the diagram – or with the crankshaft 1204 in the case of an engaged shift clutch 1205. Synchronizing devices 1251, 1252 can be provided on both shift clutches. The connection of the stator 1211 with the crankshaft 1204 with an engaged shift clutch 1205 enables operation of the electrical machine 1210 with a differential rotational speed to the rotational speed of the crankshaft 1204 so that, for example, very small differential rotational speeds can be controlled very well, and adjustment of the synchronizing rotational speeds on the transmission input shaft 1202a can occur better, more quickly, and more easily with regard to the shifting processes of the shift clutches 1205a, 1205b, 1230a.

[00205] The internal combustion engine is preferably started in a neutral position of the transmission 1201 with a disengaged shift clutch 1205a and with stator 1211 connected to the gear box casing 1250. During operation of the electrical machine 1210, the rotor 1209 accelerates the transmission input shaft 1202a, and through the gear pair 1253/1254 also the crankshaft 1204, and starts the internal combustion engine. An alternative starting method can be provided in such a manner that the shift clutch 1205a is engaged, the stator 1211 is connected with the crankshaft 1204, and a gear VI or IV that is arranged on the transmission input shaft 1202a is engaged through the shift clutch 1230a. When actuating the brakes of the vehicle, the rotor 1209 is held stationary and the stator 1211 drives the crankshaft 1204 through the gear connection 1253/1254.

[00206] Further operation of the vehicle occurs as in the examples with two friction clutches, wherein when shifting the shift clutches 1205a, 1205b the electrical machine 1210 supports or completely assumes their synchronization by accelerating and delaying the transmission input shaft 1202a with the crankshaft 1204, respectively, so that the synchronizing devices 1251, 1252 can be eliminated.

[00207] On the end of the transmission input shaft 1202a that is opposite the end with the shift clutch 1205a an auxiliary unit 1255 can be connected with it or be arranged in a connected manner, which can be driven, i.e., continued to be driven, through the electrical machine 1210 when the vehicle is stopped. Of course, additional auxiliary units can be arranged in operative connection with the transmission input shaft 1202a, which can be connected, for example, with the auxiliary unit 1255 through a continuously variable transmission.

[00208] Figure 21 shows in a diagrammatic view an example of a double clutch transmission 301, which is similar to the embodiment of the double clutch transmission 1 of Figure 1, with two transmission input shafts 302a, 302b, as well as at least one output shaft 303, which is connected with at least one driving wheel, preferably two or four driving wheels, from a drive point of view through a differential, a torque-splitting mechanism, such as a viscous-type clutch, a torque-splitting transmission, and/or the like, and thus transmits the drive torque to at least one driving wheel for the purpose of moving the vehicle, wherein a torque that is supplied by the wheels for recouperation can also be fed to the transmission in the opposite torque direction. Between the crankshaft 304 that is driven by an internal combustion engine and the transmission input shafts 302a, 302b a friction clutch 305, 306, respectively, is provided, which provides for the appropriate transmission input shaft 302a, 302b to be uncoupled from the crankshaft 304. During the transmission of the torque between the crankshaft 304 and the clutches 305, 306 a damping device for damping torsional vibrations, and/or axial or wobble vibrations, can optionally be provided, for example, a two-mass flywheel 307 that is arranged between two crankshaft branches 304, 304a or a torsional vibration damping device in at least one of the clutch disks of the clutches 305, 306. Of course, the two-mass flywheel – as is known – can be integrated into at least one, preferably both clutches 305, 306, wherein in a preferred embodiment a two-mass flywheel with double clutch can prove particularly beneficial.

[00209] The clutches 305, 306 are preferably formed as friction clutches with a pressing plate, and a pressure plate that is connected with the pressing plate in an

axially displaceable, non-rotatable manner. In particular applications, wet clutches, for example, in a multiple-disk design or similar to converter bridging clutches of torque converters, can also be beneficial, which can be integrated into the transmission. Of course, all advantages with regard to the design of converter bridging clutches, such as, e.g., profiled friction linings, piston controls for the piston controlling the converter bridging clutch, friction lining cooling, and the like, can be beneficial. When employing friction clutches, friction linings are provided axially between the pressure plate and pressing plate, which are fastened to a clutch disk that is non-rotatably connected with the respective transmission input shaft 302a, 302b. The frictional engagement between the pressure plate and pressing plate on one hand, and the friction linings on the other hand, is preferably accomplished through an axially displaceable energy accumulator, e.g. a disk spring, that applies axial pressure onto the pressing plate and pressure plate, and which is actuated preferably axially through a disengaging device, wherein the prestress between the pressing plate, the friction linings, and the pressure plate, and thus frictional engagement between the crankshaft 304 and transmission input shaft 302a, 302b, is eliminated with a disengaged clutch. Of course, when employing a double clutch, a pressing plate can be provided for both clutches 305, 306, and a disengaging device can actuate both clutches, and slipping clutch modes between an engaged and disengaged clutch can be adjusted with reduced transmittable torque. With regard to a double clutch that can be used, a self-adjusting clutch can also be provided, which is explained in detail and described in DE 100 17 815.4, which is hereby included in the present application with its entire content.

[00210] At least one disengaging device can be actuated automatically through an actuator. The actuator can be operated electrically, hydraulically, pneumatically, or in a combination of these methods, wherein, for example, an electrical actuator can supply a master cylinder, which transmits the actuation impulse through a hydraulic line to a slave cylinder, which displaces the disk spring axially by an intermediate release bearing. Additionally, a rotational drive electrical actuator can be arranged directly around the transmission input shaft 302a, 302b as an axial drive, wherein one or two axial drives, which can be nested within each other, can actuate the clutches 305, 306.

[00211] Between the transmission input shafts 302a, 302b and the transmission output shaft 303, the gears or gear ratio steps I, II, III, IV, V, VI, R are provided for forming transmission 301 with six forward and one reverse gear, wherein these are arranged on the transmission input shafts 302a, 302b in an alternating fashion with regard to their gear ratios. The reverse gear R is arranged on the transmission input shaft 302b in the depicted example, but can also be provided on the transmission input shaft 302a, e.g., adjacent to the gear ratio step VI, in other embodiments. This arrangement of the gear ratio steps results in a shifting process of the gears in such a way that, for example, a gear I can be engaged on the transmission input shaft 302b, the clutch 306 can be engaged and the next gear II can already be engaged with a disengaged clutch 305 while driving the vehicle via the transmission input shaft 302b and the transmission output shaft 303 with the gear I, and that only the clutch 305 is engaged and the clutch 306 is disengaged at the time of shifting without tractive force interruption. In order to

increase driving comfort, for example, the clutches 305, 306 can be connected in an overlapping shifting manner, i.e., that in a certain operating range both clutches 305, 306 transmit torque from the internal combustion engine to the transmission output shaft 303 in a slipping operating mode.

[00212] Based on the inventive idea, an electrical machine 310 is drivingly connected with the transmission input shaft 302a or is arranged around it in a connectable manner. In the examples illustrated, the rotor 309 with the rotor shaft 309a is arranged radially within the stator 311, whose housing is firmly connected with the transmission housing or with another stationary component.

[00213] In the diagrammatic view, the crankshaft branch 304a – as shown– splits through a connection forming a positive lock, such as a geared connection, with a gear 304b that is coaxial to the crankshaft branch 304a and two gears 304c, 304d meshing with it, which are each arranged coaxially on an input branch 304e, 304f for the clutches 305, 306 of the transmission input shafts 302a, 302b, wherein between the gears 304b, 304c, or 304b, 304d, the gear ratio i=1 or a gear ratio differing from i=1 can be adjusted, and wherein also the gear ratios i between the gears 304b, 304c and the gears 304b, 304d can be different and thus have a different gear ratio (multiplication or reduction) between the transmission input shafts 302a, 302b. Of course, the arrangement of the shafts 302a, 302b, 303 in one plane as shown here does not necessarily prove beneficial for all transmissions of this type, but rather the shafts may require a smaller space in a spatial arrangement with each other. Furthermore the transmission input shafts 302a, 302b can be designed as shafts that are arranged around each other, wherein one

transmission input shaft 302a, 302b is designed as a hollow shaft in which the other one is guided. The two clutches 305, 306 separate the transmission input shafts 302a, 302b from the crankshaft 304 and thus restrict the torque connection to and from the internal combustion engine in the disengaged state.

[00214] On the transmission input shaft 302b the idlers 312, 313, 314, 315 are rotatably arranged, starting from the clutch 306 with the smallest gear ratio or largest gear reduction (gear I) in an ascending gear ratio manner for the purpose of forming the gears I, III, V, R, and the gears are positioned through the shift sleeves or sliding sleeves 316, 317, which engage two gears I, III or V, R, respectively, by connecting one of the idlers 312, 313 or 314, 315, respectively, in the conventional manner non-rotatably with the transmission input shaft 302b, or are in a neutral position in which no gear is engaged. The idlers 312, 313, 314, 315 mesh with one of the fixed gears 318, 319, 320, 321, respectively, which are non-rotatably arranged on the output shaft 303, for the purpose of forming the gear ratios of the gears I, III, V, R, wherein for the purpose of forming the reverse gear R a reversing gear 322 meshes with both the fixed gear 321 and the idler 315. The shift sleeves 316, 317 are actuated through the end actuating mechanism 430', for example, through shift forks, which are not shown. The gear ratio steps V, R are each equipped with a synchronizing device 325, 326. On the gear ratio steps I, III the synchronizing devices are omitted. Synchronization of the transmission input shaft 302b to the rotational speed of the transmission output shaft 303 during a shifting process from the gear ratio step I to III takes place by decelerating the transmission input shaft 302b through the synchronizing device 325 of the gear ratio step V. For

this, the synchronizing device 325 is actuated and the gear III is engaged by the end actuating mechanism 430' after disengaging the gear I. The spatial sequence of the gear ratio steps I, III, V can be selected in accordance with the invention in such a way that the disengagement of the gear ratio step I and the engagement of the gear ratio step III takes place in the same axial direction of motion of the end actuating mechanism 430' as the deceleration of the synchronizing device 325. Of course, the transmission input shaft 302b can also be synchronized accordingly during a switch from the gear ratio step II into the gear ratio step IV through an appropriate end actuating mechanism, wherein here an appropriate synchronizing device should be provided on the gear ratio step VI. The electrical machine 310, as the unit that takes over the synchronization of the transmission input shaft 302a, could then – but would not necessarily have to – be eliminated or could be arranged around the transmission input shaft 302b. Furthermore, the reverse gear could also be associated with the sliding sleeve 331, which engages the gear VI, wherein the sliding sleeve 317 only engages the gear V or an additional gear VII, which has a higher gear ratio than the gear VI and on which the synchronizing device in accordance with the invention would need to be installed.

[00215] On the end opposite the clutch 305 the electrical machine 310 is connected with the transmission input shaft 302a through its rotor shaft 309a through a positive lock in the circumferential direction, e.g., it is flanged, has axial teeth, or the like. The electrical machine 310 can be arranged outside the transmission housing, wherein the rotor shaft 309a or the transmission input shaft 302a, which extends outward, is sealed against the housing. Alternatively, the

electrical machine 310 can be incorporated within the transmission housing, wherein it may prove beneficial to enclose it separately.

[00216] Additionally, the even gear ratio steps or gears II, IV, VI are arranged on the transmission input shaft 302a, wherein the gear II is arranged between gear I and III with regard to its gear ratio, gear IV between gear III and gear V, and gear VI as overdrive with the largest gear ratio. For the formation of the gears II, IV, VI the idlers 327, 328, 329 are rotatably arranged on the transmission input shaft 302a and can be non-rotatably connected with the transmission input shaft 302a through the shift sleeves 330, 331, that also are controlled by the end actuation mechanism 4301, similarly to the shift sleeves 316, 317, wherein the shift sleeve 330 can either shift one of the two gears II or IV or can be in a neutral position, in which neither of the two gears II, IV is engaged and the shift sleeve 331 engages the gear VI with the highest gear ratio or is in a neutral position. The idlers 327, 328, 329 mesh with the same fixed gears 318, 319, 320 as the idlers 312, 313, 314 of the transmission input shaft 302b and are preferably actuated with the same end actuating mechanism 430' as the gears I, III, V. The gears II, IV, VI can be synchronized in the same manner as the gears I, III, IV of the transmission input shaft 302b through synchronizing devices (not shown). Alternatively, they can be eliminated, wherein a synchronization of the idlers 327, 328, 329, which are coupled to the rotational speed of the transmission output shaft 303 through the fixed gears 318, 319, 320, occurs through the electrical machine 310, which drives or decelerates the transmission input shaft 302a accordingly to achieve the synchronizing rotational speed. During down-shifting processes, for example, from the gear ratio step III to

II or from IV to II, the appropriate transmission input shaft 302a, 302b can be accelerated through a torque that is fed by the internal combustion engine by briefly engaging the appropriate clutch 305, 306.

[00217] The shift sleeves 316, 317, 330, 331 are actuated through appropriate shift forks (not shown), which slide them axially along the transmission input shafts 302a, 302b. Actuation of the shift forks occurs automatically through one or several actuators (also not shown), for example, through electric motors and/or electric, hydraulic, and/or pneumatic valves that select appropriate kinematics – such as the end actuating mechanism 430', for example. It may prove beneficial to use an actuator not for every sliding sleeve, but one actuator for the selection motion for the purpose of selecting a shift fork for a sliding sleeve 330, 331 or 316, 317, respectively, and another actuator for the shifting motion of the selected shift fork and thus the shift sleeve. Furthermore, it may also prove beneficial to combine the two selection actuators and the two shift actuators into one actuator, respectively, wherein the inventive idea provides for a gear to become engaged on the one transmission input shaft 302a, 302b without disengaging an engaged gear on the other transmission input shaft 302b, 302a, which is also activated in the same shifting and selection arrangement.

[00218] Another beneficial embodiment can be an axial drive with an electrical rotational drive, which is arranged around the shift sleeves 316, 317, 330, 331 and thus does not require any additional devices for the transmission of motion, such as rods and the like devices. Such an axial drive is described in connection with Figure 23 of the German application having application number DE 100 15 205.8,

which is hereby included into the present application in its entire content. Finally, the usage of an end actuating mechanism 430' with one main actuating and at least one auxiliary actuating element may prove particularly beneficial, as the one further illustrated in the Figures 23 to 35c.

[00219] The function of the double-clutch transmission 301 is explained with examples based on typical operating modes such as cold starting and warm starting of the internal combustion engine, a typical upshifting process, a typical down-shifting process, up- and down-shifting process of gears arranged on a transmission input shaft 302a, 302b, support function of the drive through the electrical machine 310, sole driving with the electrical machine 310, generator function of the electrical machine 310, recouperation.

[00220] A cold starting process, for example, with outside temperatures of below 0° C, can be performed with an impulse start in this embodiment. For this, when forward driving motion is desired, initially both clutches 305, 306 can be disengaged and the sliding sleeves 317, 330, 331 in the neutral position. The shift sleeve 316 non-rotatably connects the idler 312 of the gear I with the transmission input shaft 302b, and the first gear, i.e., gear I, is engaged. The electrical machine 310 is energized and reaches the specified impulse rotational speed, e.g., 2000 to 6500 r.p.m. The impulse rotational speed can be adjusted in a variable or fixed manner in dependence on the engine characteristics such as compression, displacement, number of cylinders, and/or the like, outside temperature, oil temperature, rest period of the vehicle, viscosity of the engine and/or transmission oil, and/or the like. The clutch 305 is engaged and the internal combustion engine

is started. Immediately after the start, the clutch 306 is engaged and the vehicle starts to move. The electrical machine 310 then operates as a generator, the electrical energy that is generated is passed on to an electrical energy storage device, such as an accumulator, a high current battery, a power capacitor, and/or the like. Beneficial versions can include combinations of this with appropriate power electronics that is designed to store electrical energy over an extended period of time in a particularly effective manner, and to absorb high energy density with high efficiency quickly in a short-term storage device and release it just as quickly again. For this, particularly energy accumulation methods that use physical energy effects such as charge distribution, the build-up of electro-magnetic fields, and the like are suited, while for the long-term storage of electrical energy electro-chemical material converters, such as accumulators, batteries or the like, can be used advantageously, wherein an energy exchange can be controlled or excluded through appropriate, e.g., diode-like, circuits for different charging modes and voltages.

[00221] In a warm starting process in a warmed-up state, or at outside temperatures e.g. of above 0°C, acceleration of the electrical machine 310 to the impulse rotational speed can be eliminated and it can be started directly with engaged clutch 305. This way, a considerably quicker start of the internal combustion engine can be accomplished. Of course, with a more powerful design of the electrical machine 310, e.g., depending on the size of the internal combustion engine, at a torque of 100 Nm to 250 Nm an impulse start can be dispensed with, wherein a selection of the torque in dependence on the vehicle size and weight

between 80 and 200 Nm has proven particularly beneficial for the efficient use of the electrical machine 310 as a starter-generator, with the utilization of recouperation as well as the supportive and short-term sole operation of the vehicle.

[00222] As soon as the vehicle has started to move, e.g. in gear I, the clutch 305 is disengaged and the gear II is engaged through the shift sleeve 330. In order to activate the gear during an appropriate driving situation, for example when reaching a certain rotational speed of the internal combustion engine, the clutch 305 is engaged and clutch 306 is disengaged. Similarly, the subsequent gears III to VI are engaged by having the subsequent gear already engaged with a disengaged clutch 305 or 306, and then activating it through a torque change from one transmission input shaft to the other by engaging the one clutch and disengaging the other clutch 305, 306. Down-shifting takes places in the reverse order. Selection of a subsequent gear can occur by evaluating the driving situation, such as the speed, the acceleration, the direction of acceleration, the rotational speed of the transmission input shafts, the rotational speed of the transmission output shaft, of the driving wheels, of the non-driven wheels, the cross-acceleration, the fuel consumption, the gas pedal position, the load of the vehicle, a trailing load, and/or the like parameters. For this it may prove beneficial to integrate a control device for the transmission 301 into an overall control device of the vehicle or to connect it with the vehicle, and to evaluate the measurement parameters and characteristic lines of additional vehicle components, such as sensor signals, characteristic lines of the internal combustion engine, auxiliary units, brake system, fuel supply system, and/or the like.

[00223] In certain driving situations it may prove beneficial to perform up- and down-shifting processes where one gear that is currently used, and a desired gear that is supposed to be engaged are arranged on the same transmission input shaft 302a, such as when shifting from gear II to gear IV, or from gear IV to gear VI. For this purpose, the shift from gear II to gear IV on the transmission input shaft 302a is explained in more detail. After acceleration of the vehicle in gear II the clutch 305 is disengaged and in the meantime the clutch 306 with the engaged gear III is engaged, which allows the rotational speed of the internal combustion engine to be adjusted to the gear III and is thus lowered. The transmission input shaft 302a, which in extreme cases can rotate at the nominal rotational speed of the internal combustion engine, must be decelerated to the new synchronous rotational speed for the gear IV. So as not to have to design possibly existing synchronous rings in an oversized manner, or so as to avoid long synchronizing times in the planned synchronization by the electrical machine 310 due to the poor efficiency at which the electrical machine 310 runs at these speeds, synchronization can take place by decelerating the transmission input shaft 302a by briefly engaging the clutch 305, wherein a braking torque acting on the transmission input shaft 302a is made available by the torque of the internal combustion engine.

[00224] When down-shifting from an engaged gear to a gear on the same transmission input shaft 302a, i.e., from gear VI to gear IV, or from gear IV to gear II, for example, when the vehicle is driven at a low rotational speed of the internal combustion engine and the driver desires quick acceleration, e.g. through a kick-down actuation of the accelerator pedal, the driving torque is directed through the

transmission input shaft 302b for supplying tractive force. The procedure for this shifting mode will be explained in more detail with the example of a down-shift from gear IV to gear II. Based on the load requirement, the internal combustion engine is first accelerated to full load, and the clutch 305 is disengaged only briefly for a load-free disengagement of the shift sleeve 330 and then again partially engaged, i.e., operated in a slipping mode, so that only a portion of the torque that is made available by the internal combustion engine is directed into the clutch 305 and thus into the transmission input shaft 302a. The clutch 305 can be operated in such a way that only a specified torque is transmitted to the transmission input shaft 302a. At least one measured variable that can be used for controlling the clutch 305 can be the rotational speed of the crankshaft 304, of the transmission input shafts 302a, 302b, and/or of the transmission output shaft 303. Due to the limited supply of torque, the internal combustion engine increases its rotational speed, which allows it to reach the synchronous rotational speed for the gear III on the transmission input shaft 302a. Initially the clutch 306 is partially engaged, i.e., operated in a slipping mode, and the gear III is engaged through the shift sleeve 316, while the clutch 305 is completely engaged, wherein the internal combustion engine accelerates the transmission input shaft 302a to the new synchronous rotational speed of the gear II during optimal support by the electrical machine 310. After this synchronous speed has been reached, the clutch 306 is disengaged completely and the gear II is engaged through the shift sleeve 330.

[00225] Furthermore it may prove beneficial when starting to move the vehicle in the gear I to not engage the gear II immediately, but instead keep the clutch 305

engaged and drive the electrical machine 310 through this clutch and the transmission input shaft 302a as a generator for the purpose of generating electrical energy until the driver actuates the gas pedal. Since the acceleration process in the gear I is very brief, the synchronizing and shifting process should therefore be completed in a relatively short period of time, e.g., in less than 1 s, preferably in less than 0.5 s. For this, the transmission input shaft 302a is accelerated before starting to move the vehicle with an engaged clutch 305 through the internal combustion engine that is accelerated to full load, and after starting to move the vehicle in gear I the clutch 305 is disengaged immediately and the rotating transmission input shaft 302a is delayed to the synchronizing rotational speed of the gear II by the electrical machine 310 in the generator mode, and/or a possibly existing synchronizing device. Of course, the vehicle does not always have to start moving in gear I, particularly in the case of heavy vehicles it may be prove beneficial to start these moving with the gear II and to use the gear I only for very steep ascending slopes or as a creeping gear. In this and other cases of special versions of double clutch transmissions, it may be advantageous to provide the electrical machine on the transmission input shaft with the gear with the smallest gear ratio, for example, in this transmission 301 the electrical machine 310 is provided on the transmission input shaft 302a.

[00226] When operating the vehicle in a pulling mode, the electrical machine 310 can be operated as a generator for generating power, as already mentioned above. Furthermore, during non-pulling operation, the electrical machine 310 can recuperate, i.e., gain electrical energy during generator operation from the kinetic

energy of the vehicle, which is directed into the transmission 301 through the transmission output shaft 303. Both clutches 305, 306 can be disengaged, wherein in dependence on the speed of the vehicle a suitable gear II, IV or VI can be engaged for optimal efficiency at nominal speed of the electrical machine 310. Of course, it may prove beneficial not to uncouple the internal combustion engine in certain driving situations, for example, to use its retarding torque, particularly when the generation of electrical energy is not required, for example, with a fully charged accumulator. Furthermore, the internal combustion engine can be connected, e.g., in a slipping mode, additionally for controlling a defined recouperation torque, such as in the case of a slippery road and/or to achieve a consistent delay in ascending or descending slopes. Furthermore, in a pulling mode of the internal combustion engine, the electrical machine 10 can be operated as a generator at optimal rotational speeds near the efficiency optimum with a disengaged clutch 305 and a flow of torque via the transmission input shaft 302b through one of the gears II, IV, VI.

[00227] Figure 22 shows an example of a double-clutch transmission 401 with a modified arrangement of the gear step ratios I, II, III, IV, V, VI, R compared to the double-clutch transmission 301 of Figure 21 with basically the same function, wherein the electrical machine (not shown) can be connected with one of the two transmission input shafts 402a, 402b or one of the two transmission output shafts 403a, 403b in a coupling/uncoupling manner or in a firmly operative manner, or can be omitted, wherein the functions that are described under Figure 21 and are dependent on the electrical machine are then eliminated.

[00228] In the example shown, the transmission input shafts 402a, 402b are arranged coaxially, the transmission input shaft 402a as a hollow shaft around the transmission input shaft 402a. Guided through the clutches 405, 406, preferably outside the transmission 401 as a double clutch, the transmission input shafts 402a, 402b can preferably be connected with the crankshaft 404 in an internal combustion engine (not shown) through an intermediate damping device 407. The transmission output shaft is split in the example shown into the two branches 403a, 403b, on which the idlers 412, 413, 414, 415, or 416, 417, 418, respectively, are arranged through the sliding sleeves 438, 431, or 416a, 417a, so that they can be connected with them.

[00229] In order to form the gear sets for the individual gear ratio steps, the idlers mesh with the fixed gears 427, 428, or 432, 433, 434, which are non-rotatably arranged on the transmission input shafts 402a, 402b. The gear ratio steps I and III are arranged in operative engagement between the hollow transmission input shaft 402b and the transmission output shaft 403b, the gear ratio steps V, R with the transmission input shaft 402b and the transmission output shaft 403a. The transmission input shaft 402a is extended axially compared to the transmission input shaft 402, facing away from the clutches 405, 406, and on the axial extension the gear ratio steps IV and VI are operatively connected to the transmission output shaft 403a, and the gear ratio step II is operatively connected to the transmission output shaft 403b. Furthermore, the transmission output shaft 403b includes the parking lock P.

[00230] Synchronization of the transmission input shafts 402a, 402b, respectively, occurs during the shifting processes on a transmission input shaft 402a, 402b, respectively, through a single synchronizing device that is arranged on the appropriate transmission input shaft 402a, 402b, for example, on the gear ratio step V when shifting from the gear ratio step I to III by the synchronizing device 425, or on the gear ratio step VI when shifting from the gear ratio step II to IV by the synchronizing device 426. The torque is transmitted here by the synchronizing devices 425, 426, through the idlers 415 or 417, to the fixed gears 434 or 428 of the transmission input shafts 402a, 402b. The reverse gear R is synchronized through a synchronizing device of its own.

[00231] The shifting process of the gear ratio steps takes place through so-called end output mechanisms, parts of which are the sliding sleeves 416a, 417a, 438, 431 and shift forks (not shown) that displace the sliding sleeves axially. The end output mechanisms are actuated by an end actuating mechanism, which, in turn, is driven through an appropriate actuator system. In a beneficial version, the end actuating mechanism 430, which is only shown diagrammatically in Figure 22, has such a design that it can shift the gear ratio steps of both transmission input shafts 402a, 402b through an actuator.

[00232] Figure 23 shows, for example, an end actuating mechanism 430, as it can be used for the transmission 401 of Figure 22, and in an appropriate version that is adjusted to the transmission structure as end actuating mechanism 430' for the transmission 301 in Figure 21, as well as in a version that has been appropriately adjusted to any additional transmission with a synchronizing device on

the last gear. Of course, the subsequent end actuating mechanism can be beneficial for any additional type of transmission, e.g., additional types of load-shifting transmissions or automated transmissions with traction force interruption for minimizing the shifting times.

[00233] The example of the end actuating mechanism 430 based on the idea of the invention in Figure 23 consists of a selector shaft 462 that is actuated by an actuator system (not shown) and of the engagement means 423a, 423b, 423c, 423d. The engagement means are main actuating elements, such as shift fingers 423a, 423c, and secondary actuating elements, such as double cams 423b, 423d. The shift finger 423c is hidden by the selector shaft 462 and therefore only suggested in this figure. The engagement means interact with appropriate end output mechanisms 420, which are formed by a shift sleeve 416, 417, 438, 431 and a shift fork 465, 466, 467, 468 that is connected with it. In relation to the transmission 401 in Figure 22, the following association applies: The shift fork 465 with the sliding sleeve 416 actuates the gear ratio steps I and III, the shift fork 466 with the sliding sleeve 417 actuates the gear steps IV and VI, the shift fork 468 with the sliding sleeve 438 actuates the gear ratio steps V and R, and the shift fork 467 with the sliding sleeve 431 actuates the gear ratio step II. Furthermore the shift fork 467 can engage a gear VII (not shown) that is additionally arranged on the sliding sleeve 431, which gear could then take over the synchronizing device from gear VI as the largest gear. The shift forks 465, 466, 467, 468 are arranged on shafts 469 in an axially displaceable manner, the openings 450 of the end areas 453 of the shift forks 465, 466, 467, 468 are designed so as to interact with the respective

main actuating element such as shift fingers 423a, 423c and/or the secondary actuating element, such as double cams 423b, 423d. For this, first partial areas 451 are provided for connection with a shift finger 423a, 423c and second partial areas 452 for connection with a double cam 423b, 423d. For the purpose of engaging a gear ratio step, one of the shift fingers 423a, 423c interacts with the partial areas 451 of the appropriate shift fork 465, 466, 467 or 468 by displacing the selector shaft 462 in an axial direction, wherein the shift finger 423a interacts with a partial area 451. By rotating the selector shaft 462, the shift finger 423a, 423c turns, thus displacing the respective shift fork 465, 466, 467 or 468, in whose opening 450 the shift finger 423a or 423c – it is only one shift finger, respectively that interacts with the partial areas 451 – is located, on the shaft 469 and therefore also the appropriate sliding sleeve 416, 417, 438 or 431 is displaced and thus the appropriate gear ratio step is engaged.

[00234] At the same time, the double cams 423b, 423d have interacted with the corresponding areas 452 of all other end output mechanisms 420 that are associated with the same transmission input shaft (402a, 402b in Figure 22) during the axial displacement of the selector shaft 462, so that these gear step ratios are disengaged upon rotation of the selector shaft 462. Synchronizing a shifting process from the gear ratio step I to the gear ratio step III takes place as follows, while taking the transmission 401 of Figure 22 as an example, shifting from gear I to gear III occurs as follows: the gear that currently transmits torque is gear II, the clutch 405 is disengaged, the clutch 406 is disengaged, and gear I is still engaged. With the shift finger 423c the opening 450 of the shift fork 468 is selected and

brought into operative engagement with the partial area 451c, wherein the double cam 423b enters into operative engagement with the partial area 452a of the shift fork 465 of the gear pair I/III. A rotation of the selector shaft 462 initially causes the gear I to be disengaged due to the offset angle of the shift finger 423c to the double cam 423b, and then causes the transmission input shaft to be decelerated through the synchronizing device 425 (Figure 22) on gear V. Upon reaching the synchronizing rotational speed, or a tolerable synchronizing rotational speed range, the selector shaft 462 is rotated in the neutral position direction and is displaced axially so that the shift finger 423a operatively engages with the partial area 451 of the shift fork and is again rotated for engaging the gear III. Shifting from gear II to gear IV takes place correspondingly by first decelerating the synchronizing device 426 (Figure 22) of the gear ratio step VI through the operative engagement of shift finger 423a with the partial area 451b of the shift fork 466 and then disengaging gear II through the double cam 423d, which has an operative connection with the partial area 452d. A rotation of the selector shaft 462 effects a displacement of the sliding sleeve 466 and thus the engagement of the gear IV.

[00235] The time sequence of the engagement of the main actuating element 423a, 423c in relation to the individual secondary actuating elements 423b, 423d depends on the time-related activation of the contact between the areas 451 and the shift fingers 423a, 423c on one hand, and between the double cams 423b, 423d and the areas 452 on the other hand, during rotation of the selector shaft 462, so that an offset angle provided between the parts 423a, 423c and 423b, 423d and/or an expansion of one of the parts 423a, 423b, 423c, 423d in the circumferential

direction around the axis of the selector shaft 462 can affect the time sequence of manipulations, for example, a time delay between disengagement of the active gear ratio step, synchronization of the new gear ratio step that is to be engaged, and engagement of the new gear ratio step.

[00236] Figure 24a shows a diagrammatic view and an example of a vehicle 501, where the invention can be applied in a particularly beneficial manner. The clutch 504 in this case is arranged in the power flow between the driving motor 502 and transmission device 506; it is useful to arrange a split flywheel mass between the driving motor 502 and the clutch 504, whose partial masses can be rotated relative to each other while under intermediate control of a spring-damping device, which considerably improves particularly vibration-related properties of the drive branch. The invention is preferably combined with a damping device for absorbing or for compensating for rotational thrusts, or a device for reducing rotational thrusts, or a device for damping vibrations, as described particularly in the publications DE OS 34 18 671, DE OS 34 11 092, DE OS 34 11 239, DE OS 36 30 398, DE OS 36 28 774, and DE OS 37 21 712 by the applicant, the disclosures of which are also part of the disclosure content of the present invention.

[00237] The vehicle 501 is driven by a driving motor 502, such as an internal combustion engine, which in this case is shown as an Otto or Diesel engine; in another example, it can also be driven by a hybrid drive, an electric motor, or a hydraulic motor. The clutch 504 in the example shown is a friction clutch through which the driving motor 502 can be separated from the transmission device 506, particularly for starting to move the vehicle or performing shifting processes. By

increasing the engagement or disengagement of the clutch, more or less torque is transmitted; for this, a pressing plate and a pressure plate are shifted relative to each other in an axial manner, carrying along an intermediate friction disk. The clutch 504 is beneficially a self-adjusting clutch, i.e., the wear of the friction linings is compensated so that a consistently low disengagement force is ensured. The invention is preferably combined with a friction clutch, especially as it is described in the applications DE OS 42 39 291, DE OS 42 39 289, and DE OS 43 06 505 by the applicant, the disclosures of which are also part of the disclosure content of the present application.

[00238] The wheels 512 of the vehicle 501 are driven via a differential 510 through a shaft 508. The driven wheels 512 have associated rotational speed sensors 560, 561, wherein possibly also only one speed sensor 560 or 561 is provided, generating a signal corresponding with the rotational speed of the wheels 512; additionally, or alternatively, a sensor 552 is provided in a different suitable position in the drive train, for example, on the shaft 508, for determining the transmission output rotational speed. The transmission input rotational speed can be determined with an additional sensor, or also, as is the case in the present example, from the driving motor rotational speed, so that the gear ratio set in the transmission can be determined.

[00239] Actuation of the friction clutch 504, which is beneficially pressed, but can also be pulled in another useful embodiment, occurs in this case through an actuating device 546, such as a clutch actuator. For the purpose of actuating the transmission 506, an actuating device that includes two actuators 548 and 550 is

provided, wherein one of the actuators performs a selection operation and the other a shifting operation. The clutch actuator 546 has the design of an electro-hydraulic system, wherein an engagement or disengagement motion is generated through an electric drive, e.g., through an electric direct-current motor, and is transmitted to the disengaging system through a hydraulic line. The transmission actuators 548, 550 have the design of electric drives, e.g., electric direct-current motors, which are kinematically connected with the moving sections that are actuated for establishing the gear ratio in the transmission 506. In another example, particularly when large actuating forces are required, it may also prove very beneficial to provide a hydraulic system for actuation.

[00240] Control of the clutches 504 and the transmission 506 occurs through a control device 544, which beneficially forms a modular unit with the clutch actuator 546, wherein it may also be advantageous in another embodiment to install it in a different location in the vehicle. Actuation of the clutch 504 and transmission 506 can occur automatically in an automatic operating mode through the control unit 544, or in a manual operating mode through input by the driver through a driver input device 570, such as a shift lever, wherein the input is recorded by a sensor 571. In the automatic operating mode, gear ratio step changes are performed through appropriate control of the actuators 546, 548, and 550 based on characteristic lines, which are stored in a storage unit that is associated with the control device 544. A multitude of driving programs that are established by at least one characteristic line are available from which the driver can make his selection, such as a sporty driving program in which the driving motor 502 is operated in a

performance-optimized manner, an economy program in which the driving motor 502 is operated in a consumption-optimized manner, or a winter program in which the vehicle 501 is operated in a safety-optimized manner; furthermore, in the described example characteristic lines can be adaptively adjusted, e.g. to the driver's behavior and/or other limiting conditions, such as road surface friction, outside temperature, and the like.

[00241] A control device 518 controls the driving motor 502 by influencing the mixture or composition that is fed, wherein in the figure a representative throttle valve 522 is shown, whose opening angle is recorded by an angle transmitter 520 and whose signal is made available to the control device 518. In other examples of the driving motor control system, the control device 518 – if it concerns an internal combustion engine – receives an appropriate signal, based on which the mixture composition and/or the appropriate volume can be determined; it is also useful to use the signal of an existing lambda probe. Furthermore, in the present example the control device 518 receives a signal from a load lever 514 that is actuated by the driver and whose position is recorded by a sensor 516, a signal relating to the engine rotational speed, generated by a rotational speed sensor 528 that is associated with the engine output shaft, a signal of an intake manifold pressure sensor 526, as well as a signal of a coolant temperature sensor 524.

[00242] The control devices 518 and 544 can be incorporated in separate structural and/or functional sections; then it is useful to connect them with each other through, for example, a CAN bus 554 or another electrical connection for data exchange. It may also prove beneficial, however, to combine the areas of the

control devices, particularly since the functions cannot always be associated clearly and interaction is required. In particular, during certain phases of the gear ratio step change the control device 544 can control the driving motor 502 with regard to the rotational speed and/or the torque.

[00243] Both the clutch actuator 546 and the transmission actuators 548 and 550 generate signals from which an actuator position can at least be derived, which are available to the control device 544. The process of determining the position takes place in this example within the actuator, wherein an incremental sensor is used, which determines the actuator position in relation to a reference point. In another embodiment, it may also prove useful to arrange the sensor outside the actuator and/or to provide an absolute position determination, for example, by a potentiometer. Determining the actuator position is of particular importance with regard to the clutch actuator, because this way the engagement point of the clutch 504 can now be associated with a certain engagement branch and thus an actuator position. The engagement point of the clutch 504 is beneficially re-determined repeatedly during start-up and operation, particularly in dependence of parameters such as clutch wear, clutch temperature, and the like. Determining the transmission actuator position is important with regard to determining the engaged gear ratio condition.

[00244] Additionally, signals from rotational speed sensors 562 and 563 of the non-driven wheels 565 and 566 are available to the control device 544. In order to determine a vehicle speed, it may be useful to include the average value of the speed sensors 562 and 563, or 560 and 561, in order to compensate for speed

differences, such as when driving around curves. Through the rotational speed signals the vehicle speed can be determined, and beyond that a slippage determination can be performed. In the figure, output connections of the control devices are shown as solid lines, input connections as dotted lines. The connection of the sensors 561, 562, and 563 to the control device is only suggested.

[00245] Even on a vehicle with a drive branch as the one shown in diagrammatic view in the example in Figure 24b, the present invention can be applied particularly beneficially. On such a vehicle it is possible to change gear ratio steps without tractive force interruption. Two branches 1110 and 1120 are formed between the driving motor 1010 and output 1100, via which the torque flow can occur; each of the branches is associated with a clutch 1020 or 1030 and can be integrated through it into the torque flow. Depicted is a preferred embodiment, where the clutches 1020 and 1030 are arranged between the driving motor 1010 and gear ratio steps 1040 or 1050. In another example, it may also prove useful to arrange one or both clutches 1020 and/or 1030 between the gear ratio steps 1040, 1050 and the output 1100.

[00246] By actuating the clutches 1020 or 1030 during the change, a continuous change of the torque flow from one branch to the other can be achieved. Two groups 1040 and 1050 of gear ratio steps are available, which are included in one of the branches 1110 or 1120, respectively, wherein gear ratio steps between which an uninterrupted traction force change is supposed to be possible belong to different groups. Subsequent gear ratio steps with regard to their gear ratios preferably belong to different groups, for example, the gears I, III and V form a

group 1040 and the gears II, IV and possibly VI form a group 1050; the reverse gear R is associated with the group 1050 in a useful version. In other examples, however, it may also prove advantageous to divide the gear ratio steps in different groups, or if certain gear ratio steps can be used both in one group 1040 and in the other group 1050, or are available in both groups.

[00247] The clutches 1030 and 1020 as well as the gear ratio steps of the groups 1040 and 1050 can be actuated automatically as well, as in the example shown and described in Figure 24a. For this purpose, clutch actuators 1060 and 1070 are shown for actuating the clutches 1020 and 1030. In another example, however, it may also be very useful to use only one clutch disk for actuating both clutches.

[00248] In the figure, actuating devices 1080 and 1090 are also shown, for the purpose of actuating the gear ratio steps of the groups 1040 and 1050. One example, however, which is only equipped with one actuating device for actuating the gear ratio steps of both groups 1040 and 1050, is particularly beneficial. An actuation comprises a selection drive and a shift drive. With regard to further details of the clutch and transmission actuation as well as the control system reference is made to Figure 1a with the associated description.

[00249] Furthermore the present invention can be applied on a vehicle whose drive branch comprises a secondary branch parallel to the main branch, through which the driving torque is transmitted during a shift process in the main branch. Such transmissions have become known in various embodiments as uninterrupted traction force transmissions.

[00250] Figure 25 shows an end output mechanism with an end actuating mechanism in accordance with a particularly preferred example in accordance with the invention in an application of a vehicle as the one shown and described in Figure 24a. The end output mechanisms are formed by a coupling sleeve 601, 602, 603, 604 and a shift fork 605, 606, 607, 608, respectively, that is connected with it. A group of gear ratio steps is actuated through the end output elements 601 and 604, such as coupling sleeves; the other group of gear ratio steps is actuated through the end output elements 602 and 603. The end actuating mechanism is equipped with main and secondary actuating elements for the purpose of its connection with the end output mechanisms of both groups. A first main actuating element 611 and an additional main actuating element, which in this view is not visible, are suitable for engaging gear ratio steps; secondary actuating elements 616, 613 ensure that all other gear ratio steps of the same group, respectively, are disengaged. The shift forks 605, 606, 607, 608 are arranged on shafts 609 in an axially displaceable manner, their shift fork openings are designed so as to connect with a main actuating element, respectively, such as shift fingers 611, or a secondary actuating element, such as double cams 613, 616. For this, first partial areas 614 are provided for connection with a shift finger 611, and second partial areas 615 for connection with a double cam 613.

[00251] In order to engage a gear ratio step, for example, the shift finger 611 interacts with the end area 610 of the appropriate shift fork 605 or 606 by displacing the selector shaft 612 in the axial direction. At the same time, the double cam 613 interacts with the appropriate shift fork 607 or 608, which belongs to the same group

of gear ratio steps. A rotation of the selector shaft 612 turns the shift finger 611, thus displacing the shift fork 605 or 606 on the shaft 609, and therefore also the corresponding clutch sleeve 601 or 602, and engaging the appropriate gear ratio step. Simultaneously, the rotation of the double cam 613 causes the affected gear ratio step to be disengaged, if one was engaged.

[00252] If it is a transmission with a clutch and a drive branch, as shown in Figure 24a, secondary actuating elements interact with all other end output mechanisms, respectively, when a main actuating element interacts with a first end output mechanism. In the case of a double clutch transmission with two parallel transmission branches, secondary actuating mechanisms interact with all other end output mechanisms of a branch, respectively, when a main actuating element interacts with a first end output mechanism of this branch; this way, only one gear ratio step can be engaged in one branch at any time, however it is possible to engage one gear ratio step in each branch simultaneously.

[00253] Figures 26a, 26b, 26c, 26d show the operating mode of a secondary actuating element in more detail. Starting in Figure 26a, in which the gear ratio step belonging to the shift fork 701 is engaged and the secondary actuating element is interacting with the shift fork 701 through axial displacement of the selector shaft, the selector shaft 703 is rotated so that the end area 702 of the double cam – see 613 in Figure 25 – is pressed against the taper 701a and thus a force is generated in the disengagement direction that is larger than or equal to the required disengagement force, thus causing a disengagement motion, as shown in Figures 26b and 26c. In Figure 26d, the gear ratio step is completely disengaged, and the

selector shaft 703 can be rotated freely without power being transferred to the shift fork 701 in an engagement or a disengagement direction, wherein the double cam rotates within the circle defined by 701b. The condition shown in Figure 26d also predominates when no gear ratio step of the affected shift fork 701 has been engaged from the beginning. The secondary actuating element can be rotated freely within the circle defined by 701b.

[00254] Similar to the above described disengagement process, disengagement occurs when the other gear ratio step that has been actuated through the same shift fork is engaged. In Figure 26a, the shift fork 701 would then be displaced to the right at the beginning in relation to the selector shaft 703, and the effect would occur between the cam 702a and the taper 701c. Disengagement takes place for both gear ratio steps belonging to the shift fork 701 and for both rotational directions of the selector shaft 703.

[00255] Engagement or disengagement of an old or new gear step ratio upon rotation of the selector shaft is shown in Figure 27. First, the old gear ratio step is disengaged through the double cam, see the solid line; upon further rotation, the new gear ratio step is engaged, see the dotted line. This clarifies the closeness with regard to time at which the disengagement and engagement of the gear ratio steps, and which may even overlap slightly, which is enabled by allowing the main actuating element and the secondary actuating elements to engage simultaneously with the respective shift forks and by turning both actuating elements upon rotation of the selector shaft. The mismatch between the disengagement motion of the clutch sleeve of the old gear ratio step and the engagement motion of the new gear

ratio step is largely determined by the play of the main actuating element in the shift fork opening, the design of the double cams, and the relative angular arrangement of the main and secondary actuating elements on the selector shaft – see also Figure 28a. Particularly preferred, due to its symmetry, is an arrangement where the axis of the double cam from tip 803a to tip 803b rests perpendicular to the axis of the selector finger 802. It may however also prove useful if these axes are not perpendicular to each other, particularly when a shift fork must be actuated that shifts only one gear ratio step.

[00256] Figures 28a and 28b show an arrangement of a main actuating element 802 and a secondary actuating element 803 on a selector shaft 801. Shift fingers and accompanying double cams are located axially on the selector shaft axis at a spacing so that they interact with shift forks that are associated with the same transmission branch when the selector shaft is displaced appropriately in an axial direction, so that during a subsequent rotation of the selector shaft the affected gear ratio steps can be actuated simultaneously. From a radial point of view, the axes of the shift finger 802 and of the double cam 803 with the end areas 803a and 803b are normal to each other in the depicted preferred example.

[00257] Another arrangement is shown in Figures 29a and 29b. Two double cams 903 and 904 with their end areas 903a, 903b, 904a, and 904b are arranged next to a shift finger 902 on the selector shaft 901. In this example as well, the axes of the shift finger 902 and of the double cam 903, 904 are normal to each other. The double cams 903, 904 have a particularly wide design so that they can actuate two shift forks, respectively. This way, each of the double cams 903, 904 can

actuate two shift forks for disengaging the associated gear ratio steps. In another example, it may also prove very beneficial to combine such wide double cams with simple double cams. It may also be useful to further widen one double cam in order to actuate more than two shift forks simultaneously. The usage of particularly wide secondary actuating elements should always be preferred when end output mechanisms are to be actuated whose shift forks are located next to each other.

[00258] Figures 30a through 30c show examples of beneficial versions of secondary actuating elements. The double cam described so far is marked 'a.' Both the cam end areas and the corresponding recesses 1603 have a wedge-shaped design. As an example, one cam 1604 will be described. Depicted are two pointed, converging functional surfaces 1601a and 1601b; the cam end area 1602 is rounded. In a preferred example, the surfaces 1601a and 1701b enclose an angle of 40° to 45°, with the angle being selected accordingly larger if the force that is required to disengage the gear ratio step that is to be actuated is larger. The shape of the cam largely determines the course of the disengagement force that is generated for a disengagement motion upon rotation of the selector shaft. In another example, the shape of the cam is therefore adjusted to match the necessary force line that occurs during disengagement. The recess 1603 that corresponds to the cam encloses a slightly larger angle with the surfaces that define it than the angle of the cam. The design of the recess depends on the shape of the cam since interaction between cam and recess is limiting.

[00259] Combinations with a wedge-shaped and a rectangular-shaped corresponding part are shown in Figures 30b and 30d. In Figure 30b, the rotatable

secondary actuating element has rectangular recesses 1606, which interact with wedge-shaped cams 1607 of the displaceable shift fork; in Figure 30d, the displaceable shift fork has rectangular recesses 1608, which interact with wedge-shaped cams 1609 of the rotatable secondary actuating element. Figure 30e shows, like Figure 30a, two wedge-shaped corresponding parts, wherein here, however, the rotatable secondary actuating element 1610 has recesses 1615 and the displaceable shift fork 1611 has cams 1614. Two rectangular corresponding parts 1612, 1613 are shown in Figure 30c.

[00260] Figures 30a through 30e vary the idea of a wedge shape and a rectangular shape with a recess or a cam on the actuating element that is rotatable with the selector shaft or the displaceable end actuating mechanism.

[00261] The selector shaft position and H-shift pattern are shown in Figures 31a through 31d. The example relates to a double clutch transmission where the gear I, III, V and VII form a group that is associated with a clutch, and the gears II, IV, VI as well as the reverse gear R form another group that is associated with the other clutch. Figure 31a shows the engagement of the gear I. Since only one gear of one group can be engaged at a time, for safety, the gears III, V and VI must be disengaged when shifting into the gear I. Gear III is actuated by the same clutch as gear I, therefore it cannot be engaged at the same time in any case. Upon axial displacement of the selector shaft 1705 for the purpose of connecting the shift finger 1703 with the shift fork belonging to the gear I, the secondary actuating element 1704 interacts simultaneously with the shift fork belonging to gears V and VII. The rotation of the selector shaft 1705 for the purpose of engaging the gear I causes the

gears V or VII to be disengaged. Figure 31b shows the engagement process of gear II, where the secondary actuating element 1704 disengages the gears VI and/or R. Upon engaging gear V through the shift finger 1701, the gears I and/or III are disengaged through the auxiliary actuating element 1702, see Figure 31c.

[00262] Figure 31d shows the engagement of the gear VI, wherein the gears II and/or IV are disengaged.

[00263] Figures 32a and 32b show the function of a wide cam illustrated in Figures 29a and 29b. Upon engaging the gear II – see Figure 32a – the gears III, IV, V and/or R are simultaneously disengaged; upon engaging the reverse gear – see Figure 32b – the gears I, II, III and/or IV are simultaneously disengaged.

[00264] Figure 33a shows an example of the invention for application on a conventional automated or manually-operated transmission, which at the same time is also a particularly preferred embodiment. Although only one shift fork 1080 is shown, the described transmission has several shift forks. The shift forks 1080 of such a transmission have an engagement area 1082a for engaging a shift finger 1082b, as well as having two legs 1083a. The legs 1083a together form an arc, whose diameter at least roughly corresponds to the diameter of a bushing-shaped actuating element 1081 that is installed between the arc-shaped legs 1083a. During operation, the bushing-shaped actuating element 1081 can be rotated into certain positions, e.g., through a manually-actuated or through an actuator-operated shift rod and is axially displaceable. Through the axial displacement of the bushing-shaped actuating element 1081 a shift finger 1082b can engage with the engagement area 1082a of the desired shift fork, so that a subsequent rotation of

the bushing-shaped actuating element 1081 causes a turning motion of the shift finger 1082b and thus a displacement of the shift fork 1080. The rotation is enabled because openings 1083b, with which the leg ends 1083a can engage upon a rotational motion, are provided in the sleeve of the actuating element 1081. As already described above, additional shift forks are incorporated in the transmission at an axial distance from each other with regard to the bushing-shaped actuating element 1081. These shift forks are also equipped with arc-shaped legs, which are inserted into the bushing-shaped actuating element 1081. Since no openings like 1083b exist in the bushing-shaped actuating element 1081 axially at the height of these additional shift forks, these shift forks are fixed in their mid-position corresponding with the neutral position. This way, an actuating mechanism is connected particularly effectively with a locking device of the remaining shift forks in the neutral position for actuation of the desired shift fork. The connection of the sleeve of the actuating element 1081 with an actuating rod (not shown) takes place, for example, through bushing-shaped elements 1084. The shift finger 1082b is beneficially connected with the sleeve through a very firm connection. Particularly suited for this are welding or adhesive procedures. Alternatively, or in combination, the shift finger 1082b can be connected with the sleeve mechanically by a positive lock.

[00265] Figure 33b shows the sleeve 1090 of the actuating element 1081 more closely. In a particularly preferred version, the sleeve is made from a piece of tubing, into which openings 1091 and 1092 are incorporated, e.g., by a machining process or also through a cutting technique, such as laser cutting or flame cutting.

In their basic shape, the openings 1091 and 1092 correspond at least roughly to the cross-section of the shift fork leg 1083a, however they are slightly expanded, particularly in the circumferential direction, in order to enable the displacement of the shift fork 1080. It is also beneficial to manufacture the sleeve from flat sheet metal, which is then rolled, wherein the axial gap that is formed can remain open in the case of sufficient rigidity of the material, or it can be closed, e.g., welded. The openings 1091 and 1092 are manufactured, e.g., through punching, in the case of a flat state of the sheet metal.

[00266] Figure 34a shows an example of the invention for application on an automated transmission, as described above in further detail, which at the same time is a particularly preferred embodiment. The shift fork 1480 is equipped with a first functional area 1482a for engaging a shift finger 1482b, which is widened so much that a large enough selection passageway remains even after engagement of a gear ratio step by displacing the shift fork 1480, in order to connect with the first functional area of an additional shift fork. When a gear ratio step of this additional shift fork is engaged, the old gear ratio step should be disengaged at the same time, for which second functional areas 1483a are provided on the shift fork, which engage with the appropriate openings 1483b. Upon rotating the actuating element 1481, the shift fork is shifted into its neutral position, and the disengagement force is transmitted by the side areas of the openings 1483b, which are formed by an appropriate bent piece of sheet metal, to the wedge-shaped second functional area of the shift fork. The actuating element 1481 is formed, for example, by a bushing-shaped element 1484 and connected side elements 1485a and 1485b, preferably

made from sheet metal, and whose end areas have such a design as to form the desired functional surfaces; additionally, the shift finger 1482b is connected with the side element 1485b, wherein this connection can occur like the connection of the shift finger in Figure 33a. Figure 34a furthermore clarifies that the shift finger 1482b – the main actuating element – and the openings 1483b – the secondary actuating elements – are arranged axially relative to the axis of the actuating element 1481, and at a distance from each other in such a way that the shift finger 1482b can connect with one shift fork and the openings 1483b can connect with an additional shift fork at the same time. During a (shifting) actuation, both shift forks are actuated simultaneously so that a gear ratio step is engaged and at the same time at least one other one is disengaged, and/or it is ensured that the neutral position predominates. This figure shows only one example of a particular embodiment, the entire function of which was already described in connection with previous figures, so that representatively only one element with a main and a secondary actuating element is shown.

[00267] The side element 1485b of Figure 34a is shown in enlarged form in Figure 34b. The element is manufactured from sheet metal, preferably in a punching process. The center area 1489 is widened compared to the end areas 1486, which results in particular stability in the area of the shift finger 1482b; additionally, the end areas 1487 are easily deformable. The bent ends 1487 form the counter-piece to the second functional area 1483a of the shift fork.

[00268] The bushing-shaped element 1484 of Figure 34a is shown in more detail in Figure 34c. The element is preferably manufactured in two pieces, from a

tubular piece 1484a and a punched sheet metal collar 1484b that is connected with it, which is bent into the depicted shape through a forming process. In another example, the entire element is a single piece. The collar is then shaped in the depicted manner from a tubular piece in a forming process. The two side areas 1490 and 1491 of the openings 1483b for engaging the second functional areas 1483a of the shift fork 1480 (Figure 34a) have different designs. Only the side area 1491 is relevant for the function as the bent end area.

[00269] Figure 35a shows an example of the invention for application on a double clutch transmission, which is described in further detail above, which at the same time is a particularly preferred embodiment. The bushing-shaped element 1281 consists of two inner bushings 1285, which are arranged in such a manner relative to each other that their collars face away from each other. They carry the two side areas 1286, of which one includes a shift finger 1282b that can be operatively connected with the first functional areas 1282a. The openings or slots 1283b provided in the element 1281 are suited to connect with second functional areas 1283a in order to secure the neutral position of a shift fork, as already described above. These openings – as shown in the figure, one on each side of the shift finger – are arranged along the length of the axial sleeve of the element 1281 at a distance from the shift finger 1282b, and in such a manner that they and the shift finger 1282b can simultaneously come into contact with the desired shift forks. Recesses or notches 1284 are provided at the same axial height of the shift finger 1282b, which provides space for the second functional areas 1283a of the same shift fork during a shift movement corresponding with a rotation of the element

1281, for the purpose of actuating the fork so that an unimpeded shifting motion is enabled.

[00270] A side element 1286 of Figure 35a is shown in more detail in Figure 35b. The element is manufactured from sheet metal, preferably in a punching process. The figure shows an element with shift finger 1282b. In the flat state, the notches 1284 are punched, for example, and in a subsequent operation the element 1287 is bent into the desired radius and equipped with flanges 1290.

[00271] The patent claims submitted with the application are formulation suggestions without prejudice for achieving further patent protection. The applicant reserves the right to claim additional combinations of features that have so far only been revealed in the description and/or drawings.

[00272] References made in the dependent claims refer to the further development of the object of the main claim through features of the respective dependent claim; they should not be understood as a disclaimer for achieving independent protection of the object for combinations of features of the referred dependent claims.

[00273] Since the objects of the dependent claims can form individual and independent inventions with regard to the state of the art on the priority date, the applicant reserves the right to make them the object of independent claims or divisional declarations. Furthermore, they may also contain individual inventions which have a design that is independent from the objects of the previous dependent claims.

[00274] The exemplary embodiments should not be interpreted as a limitation to the invention. Within the framework of the present disclosure numerous changes and modifications are possible, particularly such variations, elements, and combinations, and/or materials that can be deduced by an expert with regard to the solution of the task, for example, through the combination or modification of individual features or elements or procedural steps that are described in connection with the general description and embodiments as well as the claims and contained in the drawings, and that can lead to a new object or new procedural steps or procedural step sequences through combined features, also as they relate to manufacturing, testing, and processing methods.